Chapter 10
Fluid Dynamics

10.1 Distributed Pressure Drops in Tubes and in Ducts

Pressure drops in tubes and ducts can be divided into distributed and concentrated drops. The former originate from the friction between fluid and wall. As we shall see, besides the fluid velocity, they are influenced by the viscosity, the diameter of the tube (or the actual or hydraulic diameter of the duct), and by the roughness of the wall. The latter originate from changes of the sections or direction along the path of the fluid. This means that they are located at inlet and outlet of a tube, in curves, elbows, valves, and so on. They are practically not influenced by the viscosity of the fluid and the roughness of the wall, yet they are by the velocity and by the geometric characteristics of the element disturbing the flow. Concentrated drops will be discussed in Sect. 10.2.

The pressure drop Δp along a straight pipe is computed through the following equation:

$$\Delta p = \lambda \frac{l}{d_\mathrm{i}} \rho \frac{V^2}{2}; \tag{10.1}$$

l stands for the length of the tube, d_i for the inside diameter, ρ for the density of the fluid, V for its velocity, and λ for a factor that will be specified later on. If l and d_i are in m, ρ in kg/m^3, and V in m/s, the pressure drop Δp is in Pa. Generally, it is more convenient to refer to mass velocity G.

Recalling that

$$G = V\rho, \tag{10.2}$$

and (10.1) can be written as follows:

$$\Delta p = \lambda \frac{l}{d_\mathrm{i}} \frac{G^2}{2\rho} \tag{10.3}$$

with G in kg/m^{2}s. As far as air and flue gas, it is very convenient to refer to density under normal conditions as ρ_0.

D. Annaratone, *Steam Generators*, DOI: 10.1007/978-3-540-77715-1_10,

Note that

$$\rho = \rho_0 \frac{269.5p}{T} \tag{10.4}$$

where p stands for pressure in bar and T for the absolute temperature in K.

From (10.3) based on (10.4) we obtain

$$\Delta p = 1.855\lambda \frac{l}{d_\mathrm{i}} \frac{G^2}{p\rho_0} \frac{T}{1000}. \tag{10.5}$$

In steam generators pressure p differs only slightly from atmospheric pressure. In fact, note that a relative pressure in the air or gas circuit of 6000 Pa corresponds to 0.06 bar, and absolute pressure is equal to 1.073 bar. In addition, note that the considered value is very high and can only refer to the air duct between the pusher fan and the burners of a high power pressurized generator. Generally, values are much lower. For instance, in the tubes of a smoke-tube boiler the pressure is equal to $\approx 1000\,\mathrm{Pa}$ at the most.

This explains why these modest relative pressures are frequently ignored in the computation of pressure drops. By the way, by ignoring them the value of Δp will turn out higher, thus more prudent, than the actual one. From this perspective, the only exception is represented by the ducts in depression (duct of the flue gas from the generator to the suction fan of a generator with a balanced draught). With all due reservations about ducts in depression (less frequent, in any case, given widespread pressurization), pressure p can be assimilated to atmospheric pressure assuming that $p = 1.013$ bar.

Therefore, (10.5) changes into

$$\Delta p = 1.83\lambda \frac{l}{d_\mathrm{i}} \frac{G^2}{\rho_0} \frac{T}{1000}. \tag{10.6}$$

If the calculation of Δp involves a duct with a noncircular cross-sectional area, it is required to introduce the hydraulic diameter.

By indicating the cross-sectional area of the duct with A and the wet perimeter with P (in this case it coincides with the geometric perimeter of the cross-sectional area), the hydraulic diameter is given by

$$d_\mathrm{i} = \frac{4A}{P}. \tag{10.7}$$

Thus, by indicating the sides of the rectangular cross-section with a and b, we have

$$d_\mathrm{i} = \frac{2ab}{a+b}. \tag{10.8}$$

Factor λ, called friction factor, has been the object of numerous research projects. In this book we will limit our analysis to the most significant and well-known equations.

The following is the famous equation by Blasius:

$$\lambda = 0.316\mathrm{Re}^{-0.25} \tag{10.9}$$

where Re is the Reynolds number; it is valid for $\mathrm{Re} \leq 10^5$.

The following equation by Nikuradse considers the range of the high Reynolds numbers instead:

$$\lambda = 0.032 + 0.221\mathrm{Re}^{-0.237}. \tag{10.10}$$

It is valid for $\mathrm{Re} = 10^6$–10^8.

Based on these very simple equations, it is evident that λ depends only on the Reynolds number. Further equations included later on are more complete and closer to reality because they factor in the relative roughness of the walls, as well. Relative roughness stands for the ratio between the expected higher roughness of the surface in contact with the fluid and the actual or the hydraulic diameter. By indicating it with ε Fig. 10.1 makes it possible to compute it as a function of the diameter and of the type of surface.

Karmann and Nikuradse distinguish between two boundary conditions of the flow in the pipe. The first refers to turbulent flow in a practically smooth pipe. This condition occurs when

$$\mathrm{Re}\varepsilon \leq 70 - 150. \tag{10.11}$$

In that case they suggest the following equation:

$$\frac{1}{\lambda} = 2\log_e\left(\mathrm{Re}\sqrt{\lambda}\right) - 0.8 \tag{10.12}$$

where λ is, of course, independent from ε.

The second boundary condition occurs with a perfectly rough pipe. This means that

$$\mathrm{Re}\varepsilon \geq 1000 - 2500. \tag{10.13}$$

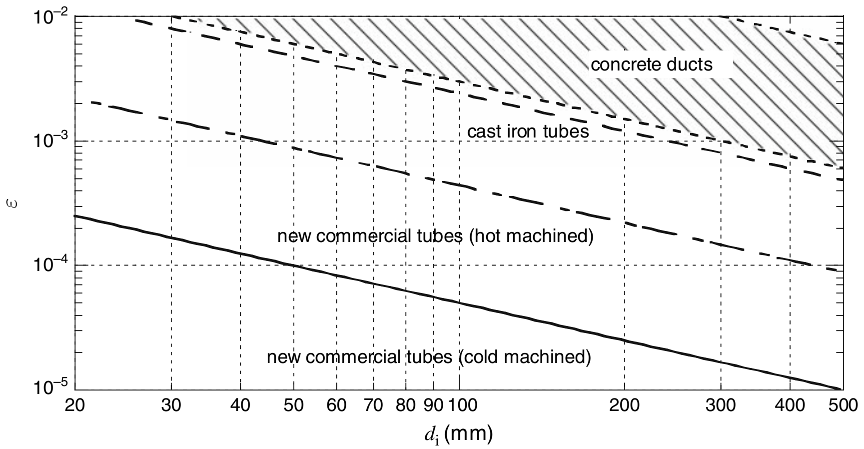


Fig. 10.1 Relative roughness ε

In that case

$$\frac{1}{\sqrt{\lambda}} = 2\log_e \frac{1}{2\varepsilon} + 1.735. \tag{10.14}$$

For values of the product Reε within 70–150 and 1000–2500, λ follows not a very well-defined rule. Karmann and Nikuradse suggested curves that are tangential to the only curve relative to the smooth pipe and to the various curves (as a function of ε) relative to the rough pipe.

Putting the equations suggested by Karmann and Nikuradse on a diagram, and following the criteria above as far as the conditions for smooth or rough pipes, we obtain a well-known diagram in the literature that we chose not to introduce in favor of Moody's diagram which is equally famous. Moody's diagram is shown in Fig. 10.2. In the case of turbulent flow (this is the only case of interest to us), the values of λ obtained from Moody's diagram can also be computed through the following equation authored by Colebrook:

$$\frac{1}{\sqrt{\lambda}} = -2\log_e \left(\frac{\varepsilon}{3.7} + \frac{2.51}{\mathrm{Re}\sqrt{\lambda}}\right). \tag{10.15}$$

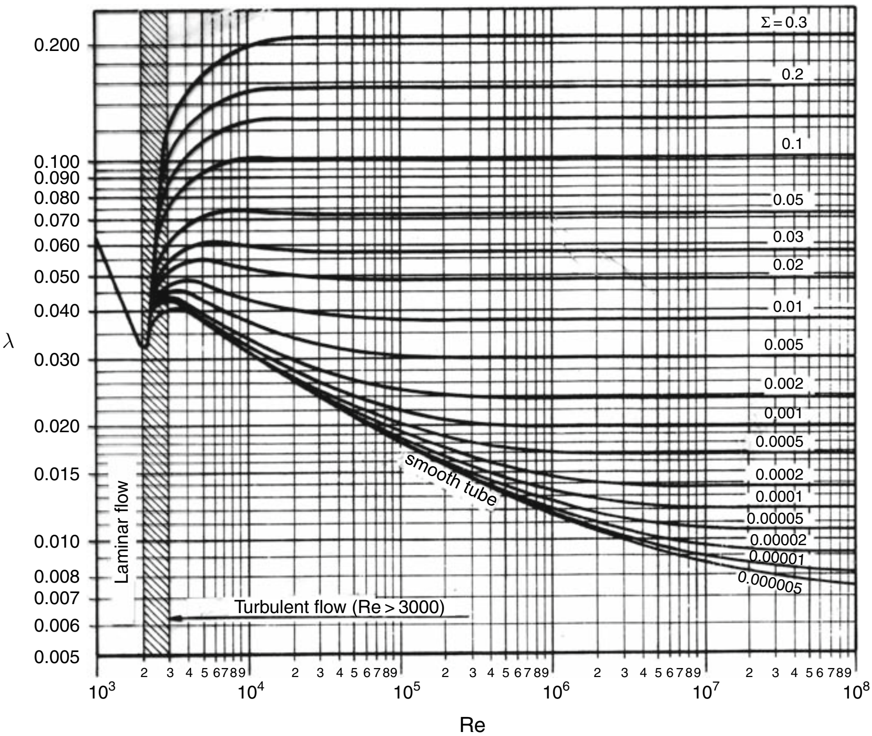


Fig. 10.2 Moody's diagram for the friction factor

The calculation of λ through (10.15) is not very straightforward because it must be done by trial and error. Still, a certain process quickly leads to a practically exact value of λ.

Note that (10.15) can be written as follows:

$$\lambda = \frac{1}{4\log_e^2\left(\frac{\varepsilon}{3.7} + \frac{2.51}{\mathrm{Re}\sqrt{\lambda}}\right)}. \tag{10.16}$$

For Re $= \infty$ from 10.16, and by indicating the value of corresponding λ with λ_0, we have

$$\lambda_0 = \frac{1}{4\left(\log_e \varepsilon - 0.568\right)^2}. \tag{10.17}$$

We write (10.16) as follows:

$$\lambda_i = \frac{1}{4\log_e^2\left(\frac{\varepsilon}{3.7} + \frac{2.51}{\mathrm{Re}\sqrt{\lambda_{(i-1)}}}\right)}. \tag{10.18}$$

This is the computation process. Given that $i = 1,\ 2, \ldots$, the values of λ_i are computed as a function of ε, Re, and λ_{i-1} until the value of λ_i practically coincides with the value of λ_{i-1} [λ_0 is obtained through (10.17)]. Convergence is quick. Further considerations can be made.

The analysis of Fig. 10.2 shows that the value of λ is influenced by the value of the Reynolds number until the latter is smaller than a certain value that depends on the value of ε.

Basically, Re does not seem to influence the value of λ when

$$\mathrm{Re}\varepsilon \geq 1000. \tag{10.19}$$

In that case the value of λ_0 computed through (10.17) diagrammed in Fig. 10.3 can be taken as the value of λ.

Equation (10.17) or the diagram can be used if (10.19) is true, or committing a mistake by defect of 3–4% at the most, if

$$\mathrm{Re}\varepsilon \geq 300. \tag{10.20}$$

Among other things, note the observations made about the equations by Karmann and Nikuradse for values ranging from 150 to 1000. According to them, the values of λ are considerably lower than those obtained from Moody's diagram in this field. In our opinion, the condition expressed in (10.20) is definitely acceptable, given the uncertainty of the values of λ for Re$\varepsilon < 1000$.

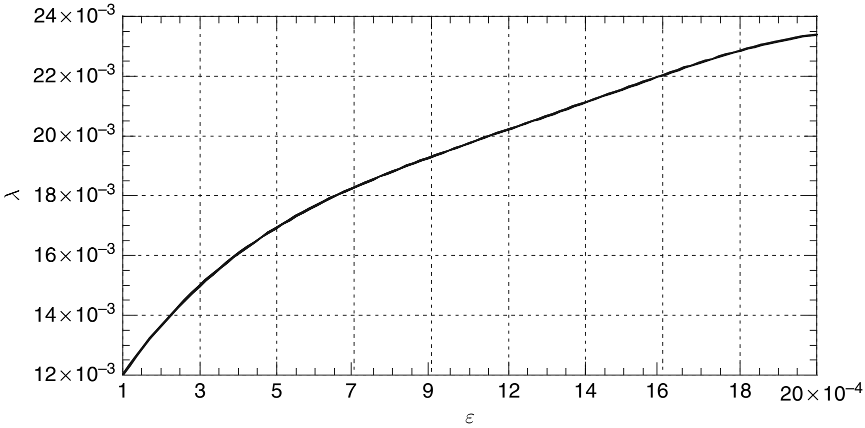


Fig. 10.3 Factor λ for a perfectly rough tube

Interestingly, for commercial warm machined steel pipes it is possible to write that

$$\varepsilon = \frac{4.4 \times 10^{-5}}{d_\mathrm{i}} \tag{10.21}$$

with d_i expressed in m.

Then, from (10.20) and recalling the significance of Re, we obtain

$$G \geq 6.8\mu \times 10^6 \tag{10.22}$$

where G is in kg/m^{2}s, whereas the dynamic viscosity μ is expressed in kg/ms.

Figure 10.4 is obtained based on the dynamic viscosity of the water. If the mass velocity of water is equal or greater than the values obtainable from this diagram as a function of temperature, it is possible to use (10.17) or the diagram in Fig. 10.3 to compute λ. Moreover, based on the values of μ, the condition expressed in (10.22) is not verified as far as the practical values of the mass velocity of air and flue gas. Note, though, that often the values of G and μ are such that the tube can be considered to be basically smooth.

If the tube is smooth, (10.15) is reduced to the following:

$$\frac{1}{\sqrt{\lambda}} = -2\log_\mathrm{e} \frac{2.51}{\mathrm{Re}\sqrt{\lambda}}. \tag{10.23}$$

The diagram in Fig. 10.5 was built based on (10.23).

The tube can be assumed to be practically smooth, and factor λ can be obtained through the diagram mentioned earlier with a mistake by defect equal to 3–4% at the most if

$$\mathrm{Re}\,\varepsilon \leq 10. \tag{10.24}$$

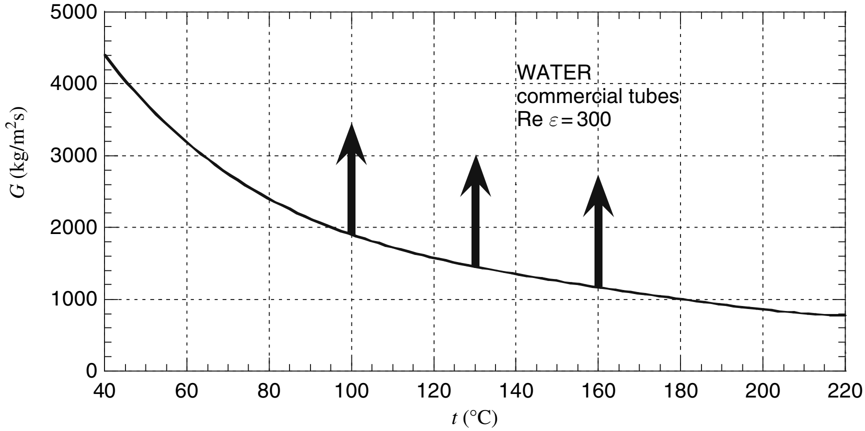


Fig. 10.4

In the case of commercial warm machined tubes, (10.24) satisfies the following condition:

$$G \leq 0.23\mu \times 10^6. \tag{10.25}$$

Figure 10.6 was built based on (10.25) and (7.91) relative to dynamic viscosity of flue gas. Figure 10.7 was built based on (10.25) and (7.78) relative to the dynamic viscosity of air. Therefore, if the mass velocity of flue gas in the tubes of the smoke-tube boiler or the air heater is smaller than the one obtainable from Fig.10.6, the value of λ can be computed through Fig. 10.5. The same is true for air flowing in the tubes of an air heater with reference to Fig. 10.7.

In terms of water and superheated steam, it is possible to refer to Mollier's tables. Nonetheless, we thought it worthwhile to show the density of water in Fig. 10.8. As

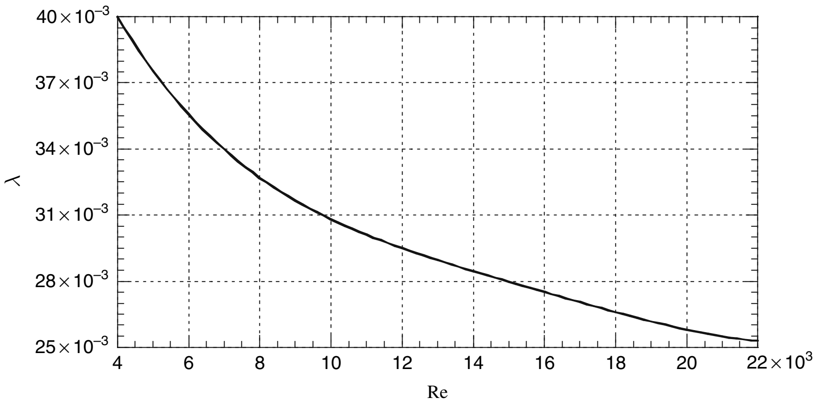


Fig. 10.5 Factor λ for a practically smooth tube

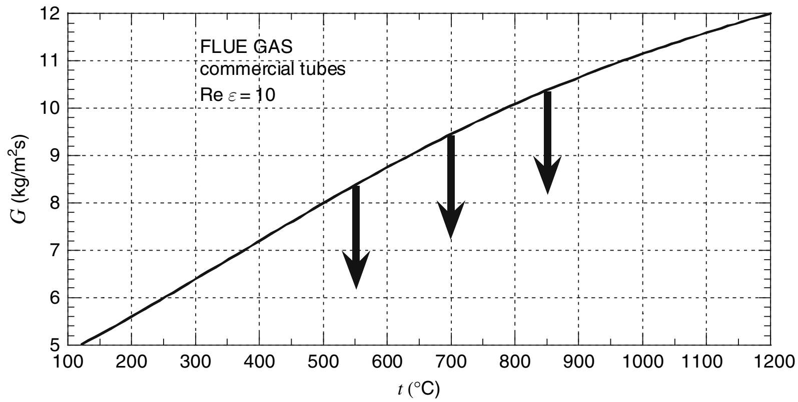


Fig. 10.6

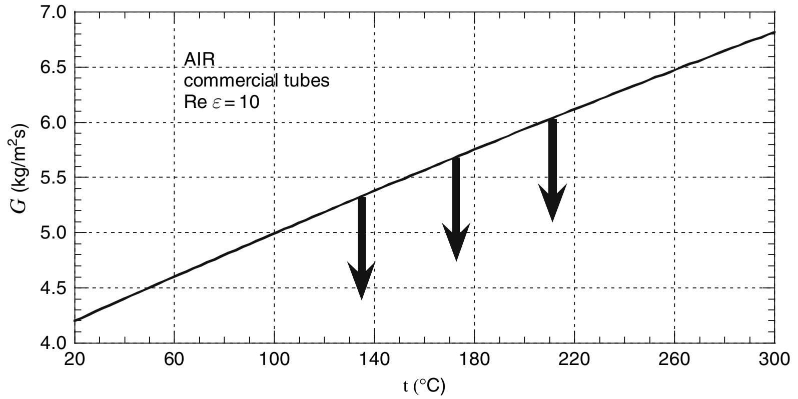


Fig. 10.7

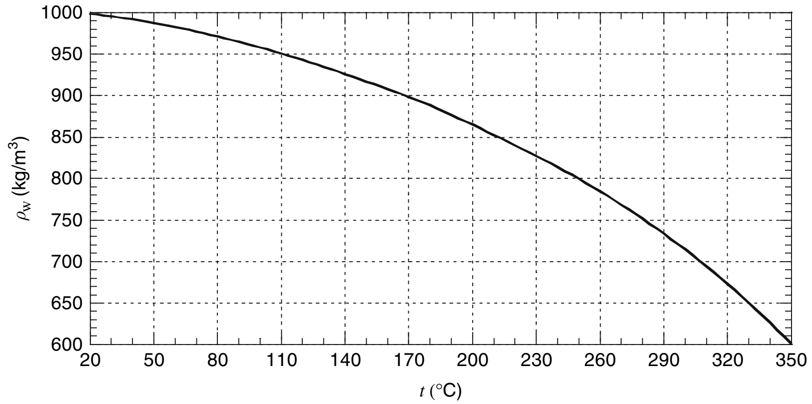


Fig. 10.8 Water density

far as flue gas and air, we already pointed out the opportunity to refer to density under normal conditions using (10.5) or (10.6) to compute Δp. The density ρ_0 of air is equal to $1.293\,\mathrm{kg/Nm^3}$. The density ρ_0 of flue gas is computed either through (7.75), or taken from Table 7.2 with sufficient approximation when the fuel is not only a mix of hydrocarbons. The value of the pressure drop Δp depends on the value of the reference temperature. In fact, the value of ρ included in (10.3) depends on this temperature. In both (10.5) and (10.6) temperature is included in an explicit way.

As far as water (economizer) or superheated steam (superheater or reheater), the average between the inlet and the outlet temperatures of the fluid from the tube bank in question can be assumed to be the reference temperature. One can proceed the same way as far as flue gas and air in air heaters. Section 10.6 illustrates the specific computation process to follow as far as steam in steam-generating tubes (mix of water and steam).

As far as flue gas flowing in the tubes of a smoke-tube boiler, it is necessary to consider the following. Because the fluid receiving the heat is at constant temperature (evaporating water), with reference to Sect. 8.12 $t_2' = t_1''$ and the factor β is zero. Thus, based on (8.34) and (8.32)

$$t_2' = t_1'' + \mathrm{e}^{-\gamma}\left(t_1' - t_1''\right) \tag{10.26}$$

where t_1' and t_2' are the inlet and outlet temperatures of the heating fluid (flue gas) and t_1'' is the inlet temperature of the heated fluid. Moreover,

$$\gamma = \frac{US_0}{M'c_p'} \tag{10.27}$$

where U stands for the overall heat transfer coefficient, M' for the mass flow rate of flue gas and c_p' for the mean specific isobaric heat, and indicating the surface of the considered passage with S_0.

Note that the presentation in Sect. 8.12 is based on U and c_p' being constant, and more specifically on the adoption of constant values for the entire temperature interval equal to the mean values. We stick to this position for now even as far as the following considerations.

Factor γ is proportional to the surface of the tubes. We may write that

$$\gamma = KS_0 \tag{10.28}$$

where K is a constant equal to $U/M'c_p'$.

Equation (10.26) can therefore be written as follows:

$$t_2' = t_1'' + \mathrm{e}^{-KS_0}\left(t_1' - t_1''\right). \tag{10.29}$$

Now, if we consider any portion of the total surface and indicate it with S, the generic temperature reached by the gas after coming in contact with surface S is given by

$$t' = t_1'' + \mathrm{e}^{-KS}\left(t_1' - t_1''\right). \tag{10.30}$$

Referring with more clarity to a numerical example with $t_1'' = 200\,°\mathrm{C}$, $t_1' = 1000\,°\mathrm{C}$ and $t_2' = 400\,°\mathrm{C}$, the pattern of the gas temperature as a function of the surface it got in contact with is represented by curve 1 in Fig. 10.9. The temperature clearly decreases with a smaller and smaller gradient along the path of the gas.

Let us examine (10.6). Given that the mass velocity is constant and that the variations in temperature have negligible influence on factor λ, we determine that the pressure drop is proportional only to the absolute temperature of the gas. Considering the pattern of the temperature of the latter, as shown in Fig. 10.9, one makes a rather big mistake by referring to the average between the inlet and the outlet temperatures of the passage. In fact, as far as Δp, this equals substitution of curve 1 with the dashed line.

Thus, it is necessary to refer to the correctly calculated mean value of the absolute temperature. To that extent note that based on (10.30) the mean value of t' is given by:

$$t_\mathrm{m}' = \frac{1}{S_0} \int_0^{S_0} \left[t_1'' + \mathrm{e}^{-KS} \left(t_1' - t_1'' \right) \right] \mathrm{d}S. \tag{10.31}$$

Then, resolving the integral

$$t_\mathrm{m}' = t_1'' - \frac{t_1' - t_1''}{KS_0} \left(\mathrm{e}^{-KS_0} - 1 \right). \tag{10.32}$$

Based on (10.29)

$$\mathrm{e}^{-KS_0} = \frac{t_2' - t_1''}{t_1' - t_1''}; \tag{10.33}$$

$$KS_0 = \log_\mathrm{e} \frac{t_1' - t_1''}{t_2' - t_1''}. \tag{10.34}$$

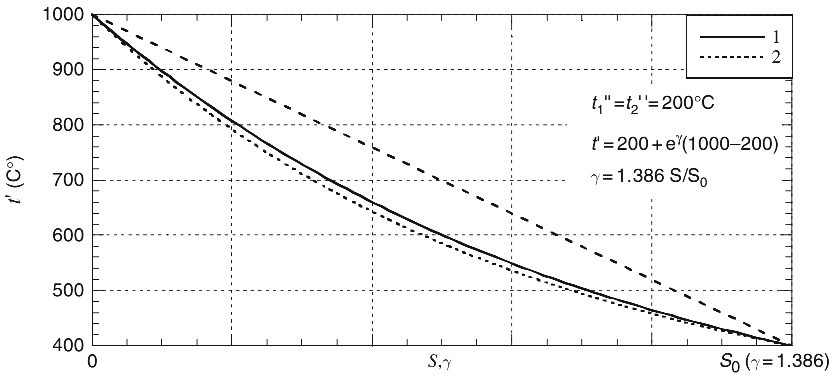


Fig. 10.9 Flue gas temperature inside smoke tubes or through a steam generating tube bank

Based on (10.32) we obtain

$$t'_{\mathrm{m}} = t''_1 - \frac{t'_1 - t''_1}{\log_e \dfrac{t'_1 - t''_1}{t'_2 - t''_1}} \left(\frac{t'_2 - t''_1}{t'_1 - t''_1} - 1 \right); \tag{10.35}$$

and after a series of steps

$$t'_{\mathrm{m}} = t''_1 + \frac{t'_1 - t'_2}{\log_e \dfrac{t'_1 - t''_1}{t'_2 - t''_1}}. \tag{10.36}$$

Equation (10.36) correctly computes the mean temperature of the gas, thus its mean absolute temperature to introduce in (10.5) or in (10.6) for the computation of Δp.

Note that (10.36) can also be written as follows:

$$t'_{\mathrm{m}} = t'_1 \left(\frac{t''_1}{t'_1} + \frac{1 - \dfrac{t'_2}{t'_1}}{\log_e \dfrac{1 - \dfrac{t''_1}{t'_1}}{\dfrac{t'_2}{t'_1} - \dfrac{t''_1}{t'_1}}} \right); \tag{10.37}$$

finally,

$$t'_{\mathrm{m}} = \varphi t'_1 \tag{10.38}$$

where the dimensionless factor ϕ which corresponds to the expression in parenthesis in (10.37) can be directly taken from the diagram in Fig. 10.10 as a function of t'_2/t'_1 and t''_1/t'_1. The superior line represents the values of ϕ corresponding to the linear

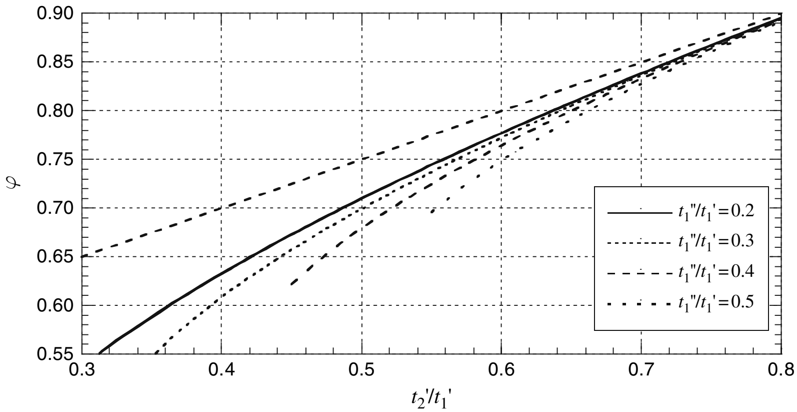


Fig. 10.10 Factor φ for the calculation of the mean flue gas temperature

pattern of the temperature (like the straight line in Fig. 10.9). This highlights the influence of the actual pattern of the temperature on the value of t'_{m}.

Of course, the mistake made by adopting the average between the extreme temperatures would be increasingly greater as the ratio between them were to farther away from one. One could argue that the computation process is invalidated by the variability of U and c'_p with the temperature. In fact, this does have a certain impact. With reference to our example, the curve that stands for the actual pattern of the temperature is represented by curve 2 in Fig. 10.9. These considerations can be applied to a steam-generating tube bank, too, as we shall be able to see in Sect. 10.3.

Thus, if it is not allowed to assimilate the theoretical curve to the dashed line in the diagram, one does not commit a sensible mistake by adopting the theoretical curve instead of the actual one. Moreover, note that the pressure drop computed this way is slightly greater than the actual one and consequently in favor of safety during runtime.

Thus, (10.38) provides good approximation to the reality of the phenomenon. Based on (10.1) and the considerations above, we determine the following. The friction factor λ depends on the type of tube or duct surface (absolute roughness). It is also a function of the diameter and, through Re, a function of the fluid velocity and its kinematic viscosity. The pressure drop per length unit of the tube is a function of the absolute roughness of the surface, of the inside diameter, of the velocity, of the dynamic viscosity, and of the density.

If R indicates absolute roughness, we have

$$\frac{\Delta p}{l} = f\left(R, V, d_{\mathrm{i}}, \mu, \rho\right). \tag{10.39}$$

On the other hand, for a given fluid μ and ρ are a function of temperature only. Thus, we can write that

$$\frac{\Delta p}{l} = f\left(R, V, d_{\mathrm{i}}, t\right). \tag{10.40}$$

We determine that once a type of surface (for instance, commercial tubes) and a reference temperature are set, the pressure drop by length unit of a given fluid is only a function of velocity and diameter.

This way it is possible to create diagrams to obtain $\Delta p/l$ directly.

The diagrams in Figs. 10.11 and 10.12 refer to commercial steel tubes and water flowing through them at 20 °C. The pressure drop referred to the length unit of the tube can be obtained based on two of the quantities Q, V, and d_i (the volumetric flow rate is expressed in m^3/h).

If the temperature is different from 20 °C, the actual pressure drop Δp can be obtained by first approximation from the following equation and by indicating the pressure drop gained from the diagrams with Δp^*, thus

$$\Delta p = \Delta p^* \left(1 - 1.04 \frac{\sqrt{t-20}}{100 V^{0.4}}\right) \frac{\rho}{1000}; \tag{10.41}$$

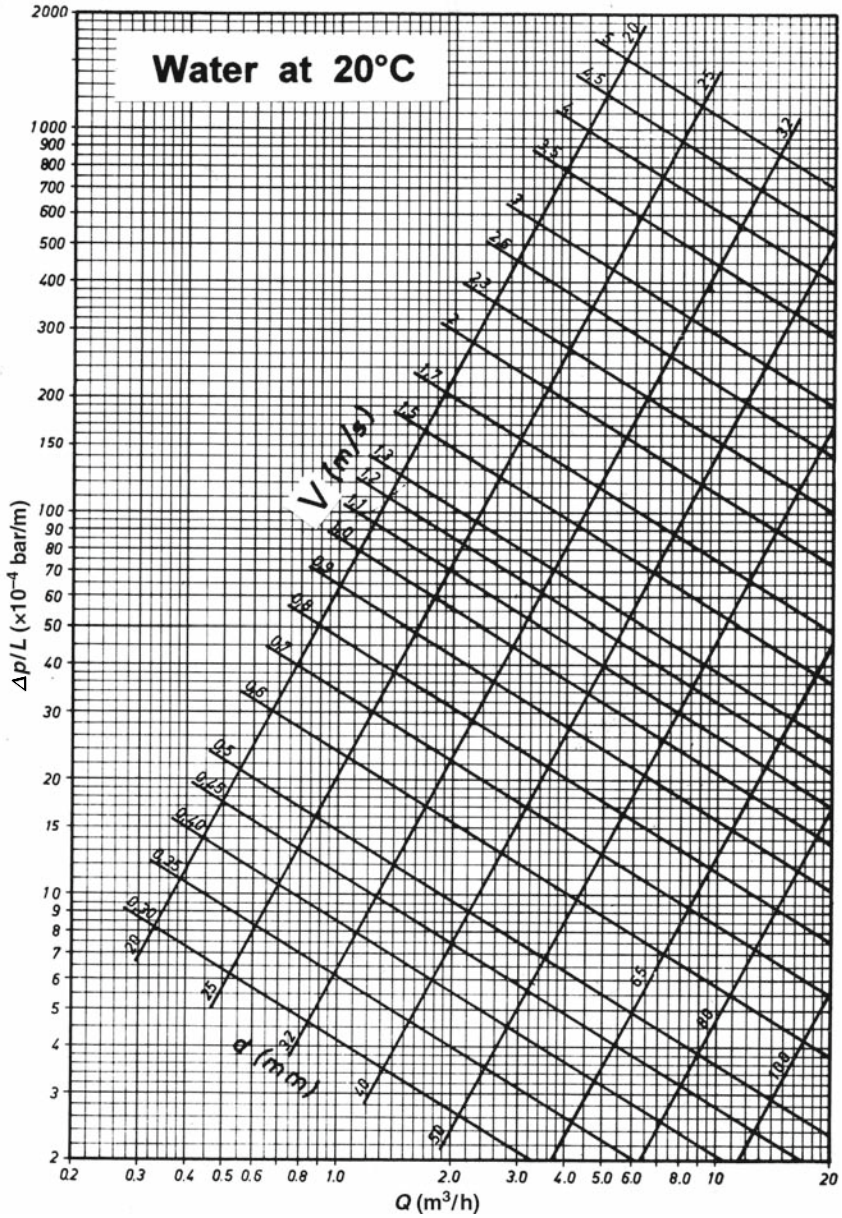


Fig. 10.11 Pressure drop for water at 20 °C ($Q = 0.2$–$20\,\mathrm{m^3/h}$)

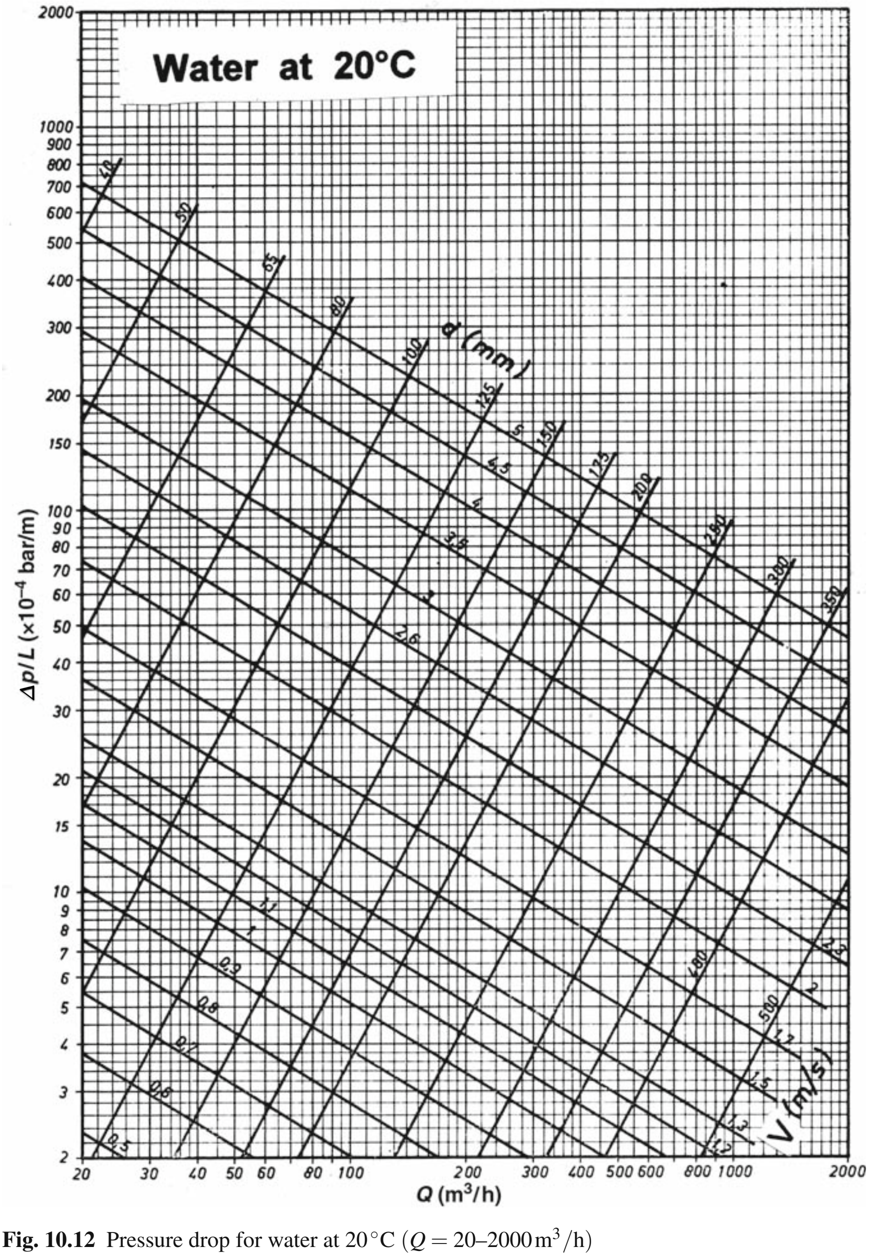


Fig. 10.12 Pressure drop for water at 20 °C ($Q = 20–2000\,\mathrm{m^3/h}$)

ρ stands for the water density in kg/m^3, V for the velocity in m/s, and t for the temperature in °C.

Fig. 10.13 as well as 10.14 show the diagrams relative to $\Delta p/l$ for commercial tubes or ducts in metal sheet with air under normal conditions.

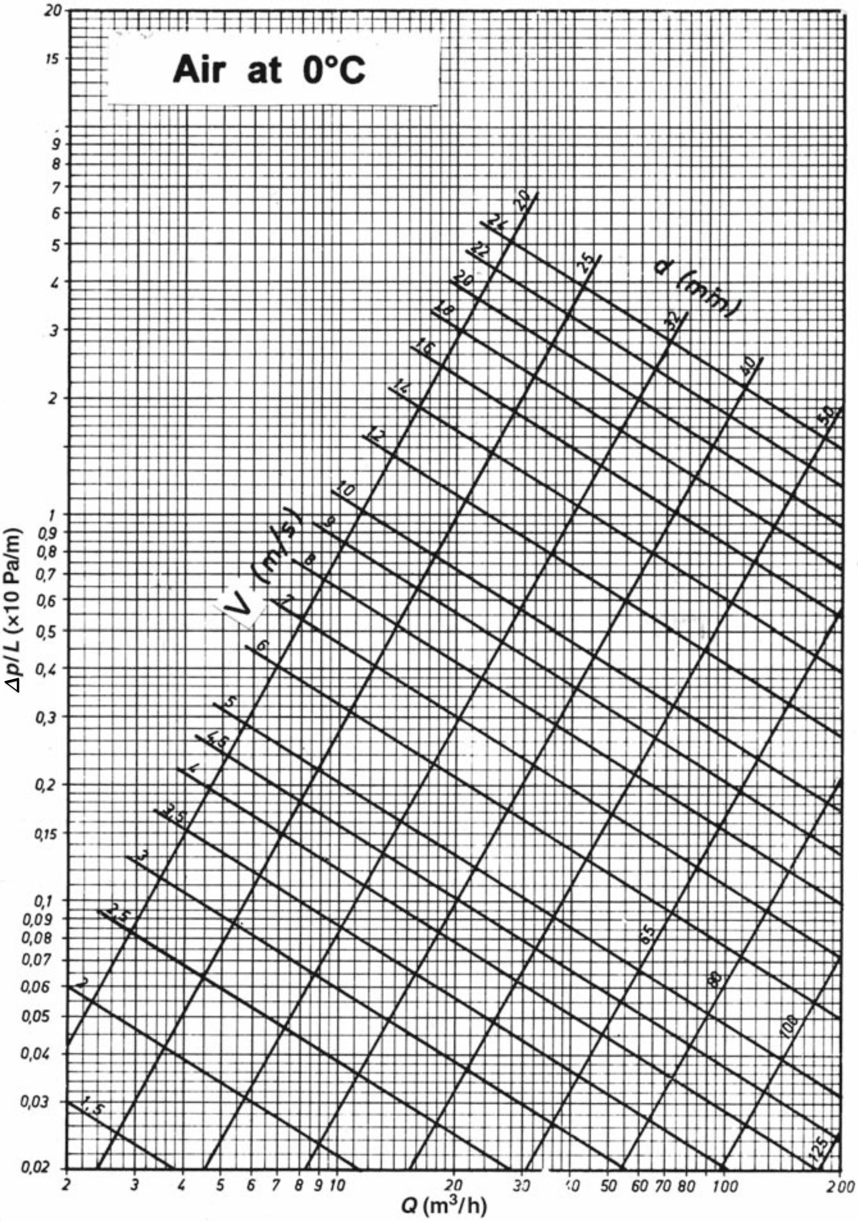


Fig. 10.13 Pressure drop for air at 0 °C ($Q = 2$–$200\,\mathrm{m}^3/\mathrm{h}$)

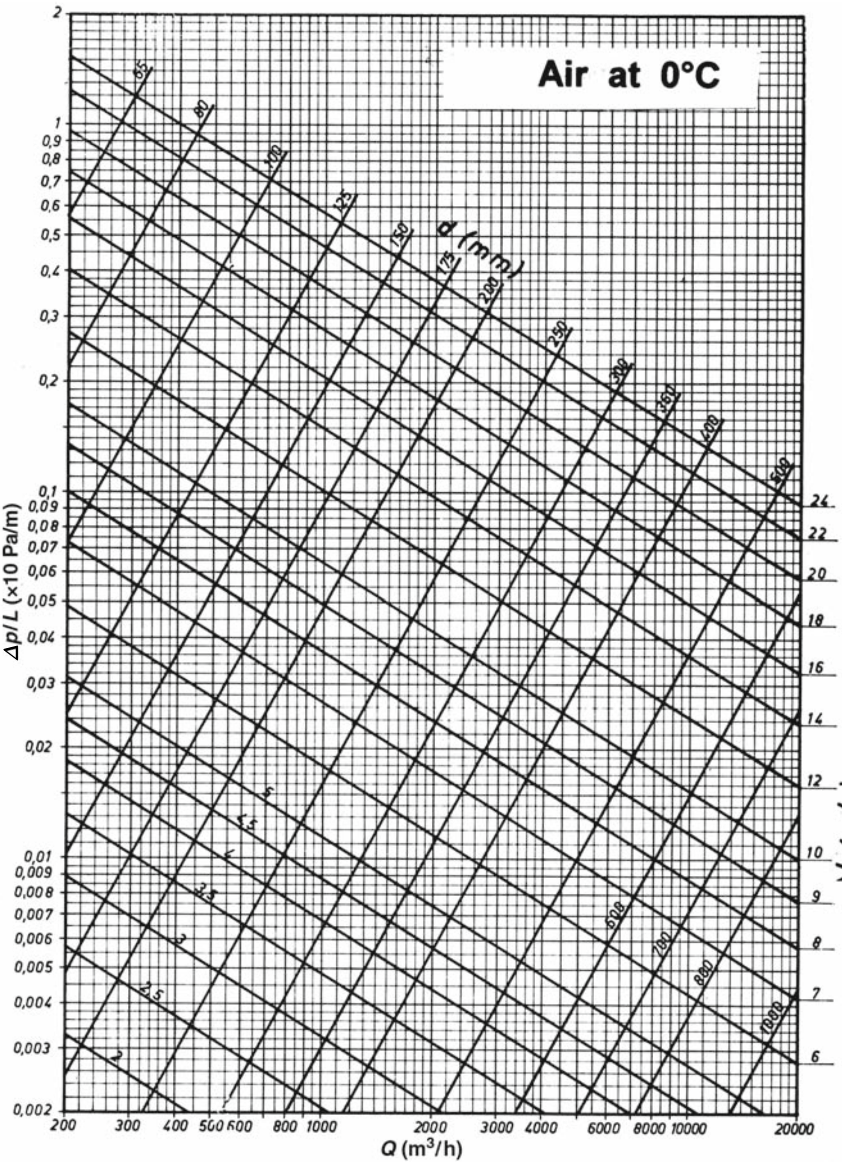


Fig. 10.14 Pressure drop for air at 0°C ($Q = 200\text{–}20{,}000\,\text{m}^3/\text{h}$)

If the temperature is different from 0°C, the actual pressure drop Δp can be obtained by first approximation from the following equation and by indicating the pressure drop gained from the diagrams with Δp^*, thus

$$\Delta p = \Delta p * \left(1 + 2.55 \frac{\sqrt{t}}{100 V^{0.6}}\right) \frac{273}{273 + t} \tag{10.42}$$

where V stands for the velocity in m/s and t for the temperature in °C.

Note that the reference to a commercial tube indicates a new tube. If the tube shows increasing roughness during runtime caused by wear and tear, the pressure drop increases. The increase is small for low values of the Reynolds number (air and flue gas), but it can be considerable for high values of Re (superheated steam and water).

Finally, note that in the case of ducts without a round cross-section, the diagrams above must be used in reference to velocity and diameter, that is, the quantities that are crucial for the value of Δp (10.40). The value of the volumetric flow rate Q on the abscissa corresponds to the values of V and d_i only if the cross-section is round, but not if d_i is the hydraulic diameter.

10.2 Concentrated Pressure Drops

As already pointed out in the previous section, concentrated pressure drops are caused by inlets and outlets, curves, changes in the cross-sections, offtakes, and so on. They are computed as follows:

$$\Delta p = \zeta \frac{\rho V^2}{2} \tag{10.43}$$

where ζ is a factor that shall be explained later on. If ρ is in $\mathrm{kg/m^3}$ and V in m/s, the pressure drop Δp is in Pa.

Similarly to distributed pressure drops, if we refer to mass velocity G expressed in $\mathrm{kg/m^2 s}$, from (10.43) we have

$$\Delta p = \zeta \frac{G^2}{2\rho}. \tag{10.44}$$

By introducing density under normal conditions ρ_0 of both air and flue gas, from (10.44), and similarly to (10.5), we obtain

$$\Delta p = 1.855\zeta \frac{G^2}{p\rho_0} \frac{T}{1000}. \tag{10.45}$$

By assimilating pressure p to atmospheric pressure (1.013 bar), from (10.45) we obtain the following equation:

$$\Delta p = 1.83\zeta \frac{G^2}{\rho_0} \frac{T}{1000}. \tag{10.46}$$

Factor ζ in the previous equations can be considered to be practically independent from the Reynolds number when the latter is greater than 3000–4000, as is the case in the instances of interest to us.

For instance, Fig. 10.15 shows curves standing for ζ for the inlet and outlet of a tube from a tank for which we adopt $\zeta = 0.5$ and $\zeta = 1$, as we shall see later on.

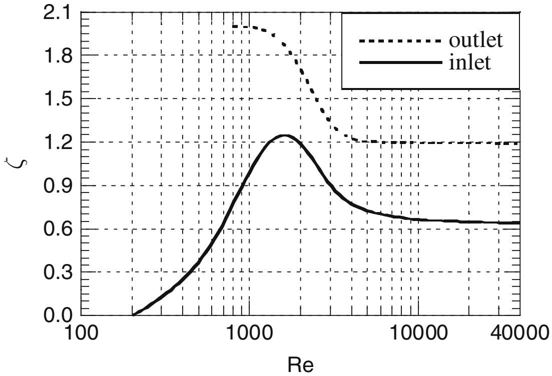


Fig. 10.15 Factor ζ for inlet and outlet in a tank

Inlet and outlet from a tank are, in fact, threshold instances of sudden sectional changes. Two cases stand out during the analysis of this issue in general terms. In the first case, the cross-section abruptly decreases along the pass of the fluid. The value of ζ is obtained from curve A of Fig. 10.16 as a function of ratio $d_\mathrm{i}/d_\mathrm{i}'$ between the diameter of the reduced cross-section and the diameter of the original cross-section. If the fluid enters the tube exiting from a tank, we may consider that $d_\mathrm{i}' = \infty$, thus $d_\mathrm{i}/d_\mathrm{i}' = 0$ which finally leads to $\zeta = 0.5$.

In the second instance the cross-section widens, the value of ζ is obtained from curve B of Fig. 10.16. If the fluid enters in a tank in a similar way, we can consider that $d_\mathrm{i}/d_\mathrm{i}' = 0$, thus $\zeta = 1$. In that case all the kinetic energy of the fluid is lost.

In both instances the velocity to consider in (10.43) is the one in the reduced cross-section. In general (here the equation can also be used with a duct with a non-circular cross-section), factor ζ for abrupt reduction of the cross section is given by

$$\zeta = 0.5\left(1 - \frac{A}{A'}\right) \tag{10.47}$$

where A stands for the smaller and A' for the greater cross-section.

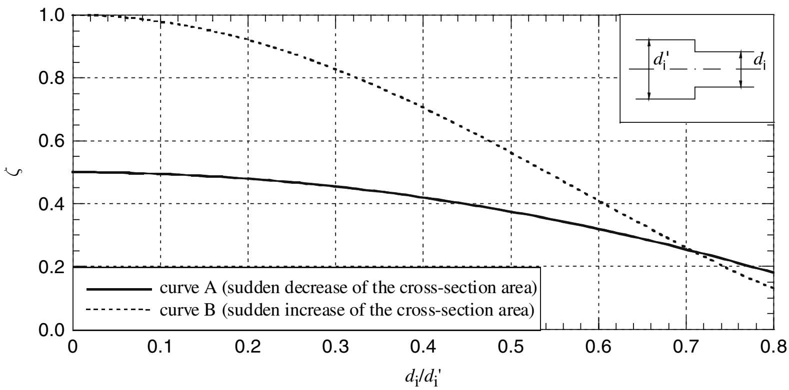
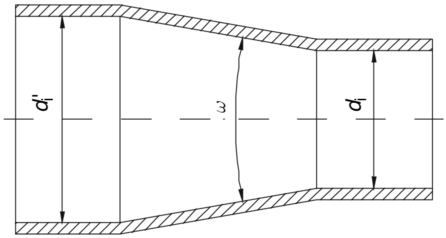


Fig. 10.16 Factor ζ for sudden variations of the cross-sectional area

Fig. 10.17



In the case of widening of the cross-section we have

$$\zeta = \left(1 - \frac{A}{A'}\right)^2, \tag{10.48}$$

including the same significance of symbols.

If the cross-section is gradually reduced with an angle ω smaller than 30° (Fig. 10.17) instead, it is possible to assume that $\zeta = 0.05$ regardless of the value of d_i/d_i' or A/A'. If the cross-section gradually widens, the values obtained from curve B in Fig. 10.16 or from (10.48) are multiplied by the corrective factor β, the values of which are shown in Table 10.1.

In the case of inlet from a tank, if the tube enters part of the tank itself (Fig. 10.18), we can assume that $\zeta = 0.8$.

As far as the pressure drop of inlet and outlet relative to tubes of smoke-tube boilers, given that flue gas has a certain kinetic energy at the tube inlet and that not all kinetic energy is lost at the outlet, it is possible to adopt the following values of ζ: for the inlet $\zeta = 0.35 - 0.4$ and for the outlet $\zeta = 0.65 - 0.7$.

Finally, it may be interesting to determine the pressure drop through a drilled diaphragm (Fig. 10.19) for the sizing of the throttling baffle plates. In that case

Table 10.1 Factor β for gradual increase of the cross-sectional area

ω	3°	5–8°	10°	14°	20°	30°	45°	60°	≥ 90°
β	0.18	0.14	0.16	0.25	0.45	0.70	0.95	1.10	1.00

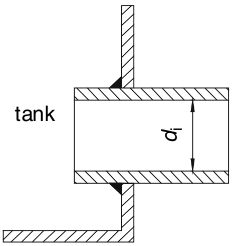
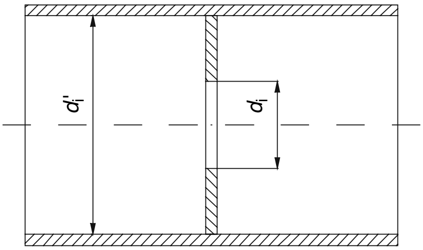


Fig. 10.18

Fig. 10.19



factor ζ is obtained with reference to curve B in Fig. 10.16 and by multiplying the value obtained this way by the corrective factor β given by

$$\beta = 2.8\left[1 - \left(\frac{d_i}{d_i'}\right)^2\right]. \tag{10.49}$$

More in general, the value of ζ computed based on (10.48) is multiplied by the corrective factor β given by

$$\beta = 2.8\left(1 - \frac{A}{A'}\right). \tag{10.50}$$

The velocity to consider in (10.43) or in the equations obtained from it is always the one referring to the reduced cross-sectional area.

Besides the bending angle, factor ζ relative to pressure drops in the curves is also a function of the ratio between the bending radius referred to the axis of the tube and its inside diameter. The diagram in Fig. 10.20 leads to its value for 45°, 90°, 135°, and 180° curves. The interpolation is done for intermediate angles. In the case of ducts it is worthwhile knowing the values relative to elbows, too (Fig. 10.21). In the absence of baffles they are taken from Table 10.2.

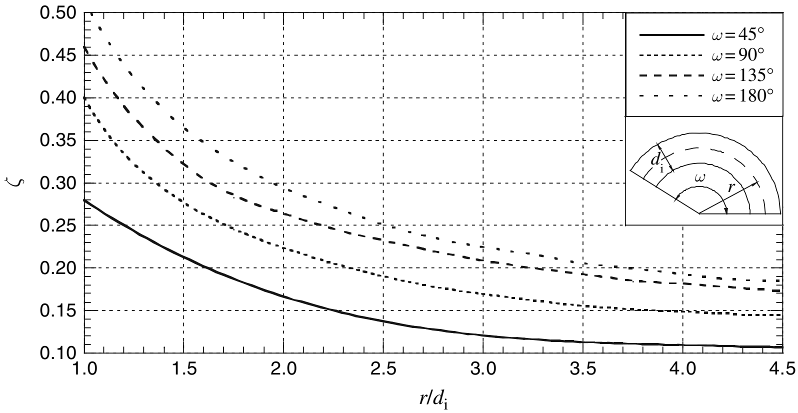


Fig. 10.20 Factor ζ for curves

Fig. 10.21

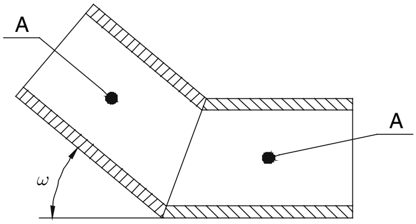


Table 10.2 Factor ζ for elbows without baffles

ω	15°	30°	45°	60°	90°
ζ	0.02	0.07	0.18	0.36	1.00

If the elbows are equipped with baffles instead, it is possible to refer to Table 10.2 by multiplying the values by 0.4.

As far as the T and Y-shaped offtakes, the values of ζ are not available for all the numerous potential instances, both in terms of the different values of volumetric flow rates in both branches of the offtake and in terms of the different diameters of the concurrent tubes. The diagram in Fig. 10.22 makes it possible to obtain ζ for T-shaped offtakes with concurrent tubes of equal diameter as a function of the ratio between the flow rates and two possible flux directions.

The diagram in Fig. 10.23, instead, makes it possible to obtain ζ for Y-shaped offtakes with concurrent tubes of equal diameter as a function of angle ω between the two spread out branches under the assumption that the flow rate in both branches is the same and for the two possible flux directions.

The velocity to consider in (10.43) and the like is the one corresponding to volumetric flow rate Q.

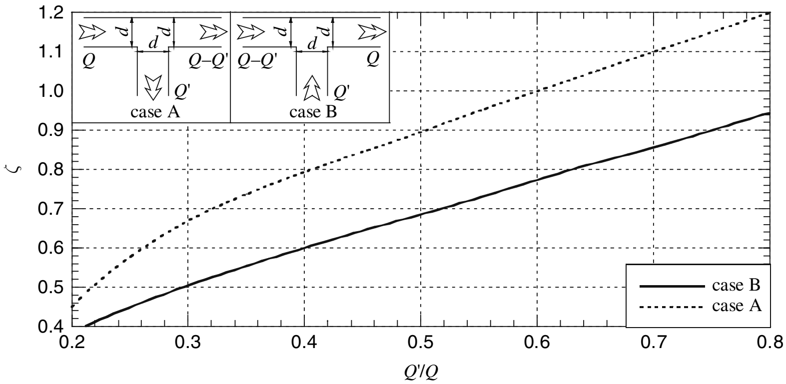


Fig. 10.22 Factor ζ for T pieces

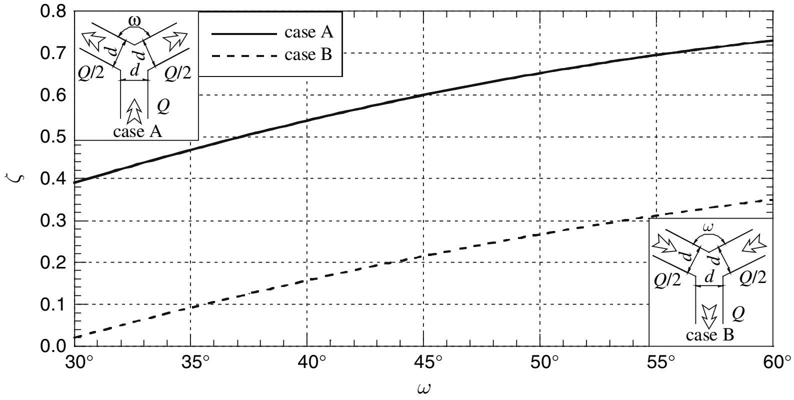


Fig. 10.23 Factor ζ for Y pieces

Other authors suggest the following values for T-shaped offtakes with equal diameters and distinguishing the two values of ζ relative to continuous piping and the one at 90° that we indicate with ζ and ζ', respectively. In case B of Fig. 10.22

$$\begin{array}{lll} \text{for } Q' = 0 & \zeta = 0.04 & \zeta' = 0 \\ \text{for } Q' = 0.5Q & \zeta = 0.40 & \zeta' = 0.30 \\ \text{for } Q' = Q & \zeta = 0 & \zeta' = 0.90 \end{array}$$

In case A of Fig. 10.22

$$\begin{array}{lll} \text{for } Q' = 0 & \zeta = 0.04 & \zeta' = 0 \\ \text{for } Q' = 0.5Q & \zeta = 0.01 & \zeta' = 0.90 \\ \text{for } Q' = Q & \zeta = 0 & \zeta' = 1.30 \end{array}$$

The values of both ζ and ζ' refer to the velocity corresponding to Q. As far as the valves, there are many more kinds. Definitely reliable data can only be obtained by consulting with the producing company. The same is true for gate valves. Finally, flow rate meters, slide regulation valves, and expansion compensators are inserted in the ducts. It is impossible to provide general information on the relative pressure drops. If the flow rate meter is a Venturi-type meter, computation is still possible by examining its shape and by applying the described values of ζ, even though in this case it is still recommended to turn to the specialist company for available experimental data on their instruments. As far as slide valves and compensators, there are experimental data, or in the lack thereof it is a question of reasonably increasing the total pressure drop in the duct.

In conclusion, note the widespread criterion to assimilate the generic concentrated drop to the one in a portion of straight tube of a certain length, called virtual

length or equivalent length. A comparison between (10.43) and (10.1) shows the requirement for equal pressure drops to be

$$\lambda \frac{l_e}{d_i} = \zeta, \tag{10.51}$$

where l_e stands for the virtual length (equivalent length).

Then

$$l_e = \frac{\zeta}{\lambda} d_i. \tag{10.52}$$

The criterion above actually consists of computing a virtual length equal to a certain number of diameters for every concentrated drop. Given that N_e is this number, from (10.52) we obtain

$$l_e = N_e d_i; \tag{10.53}$$

then

$$N_e = \frac{\zeta}{\lambda}. \tag{10.54}$$

The criterion is interesting because it simplifies the computation of the pressure drop of a tube or a duct. In fact, once the values of N_e for the different "flux perturbations" are known, one calculates a fictitious length l' given by

$$l' = l + d_i \sum N_e \tag{10.55}$$

where l stands for the length of the tube with a constant diameter d_i.

Δp is computed based on l' through one the equations in Sect. 10.1 or deriving the value of $\Delta p/l'$ directly from the diagrams. This takes both distributed and concentrated pressure drops into account. Note that to safeguard the validity of the method (i.e., to simplify computation) requires a constant value for N_e which is typical of any type of perturbation. This is what is actually done. But based on (10.54) N_e is clearly a function of λ and ζ; thus, it is influenced by the Reynolds number, as well as the roughness. Calculation based on constant values of N_e is therefore acceptable only for an approximate evaluation of Δp, especially if the concentrated pressure drops have a considerable impact on the total.

On the other hand, if the values of N_e were to be correlated to the values of ζ, Re, and ε, one would be forced to create an enormous series of diagrams for all the types of perturbations and for the different values of the quantities in question (ratios between diameters, curvature radii, and so on). At this point, the usefulness of the method would be gone. In other words, this easy calculation criterion can be recommended only for gross evaluations of Δp. These reservations notwithstanding, it can be interesting to determine some criteria to obtain approximate yet constant values of N_e. For flue gas flowing in the tubes of smoke-tube boilers the values of Re usually range from 8000 to 25000. The relative roughness ranges from 5×10^{-4} to 10^{-3}. Based on Fig. 10.2, λ ranges from 0.026 to

0.034. By adopting an average value equal to 0.030, based on (10.54) we may write that

$$N_e = 33\zeta. \tag{10.56}$$

As far as the pressure drop at the inlet, considering the extreme case of an inlet from a tank, so that $\zeta = 0.5$, the virtual length corresponds to 16.5 diameters. For the outlet with $\zeta = 1$, we have $l_e = 33d_i$ instead. Actually, the pressure drops of inlet and outlet are smaller because there is a certain kinetic energy at the inlet, and at the outlet the kinetic energy of the flue gas exiting the tubes is not completely lost, as we already pointed out before.

As far as superheated steam, the conditions vary greatly depending on pressure and temperature. For pressure ranging from 15 to 100 bar and temperatures up to 500 °C, the variability range of Re from 150,000 to 1,000,000 may be considered for the tubes of superheaters; the relative roughness may be assumed to be equal to $1\text{–}2 \times 10^{-3}$; Figure 10.2 shows that λ varies from 0.020 to 0.024. Assuming an average value equal to 0.022 we have

$$N_e = 45\zeta. \tag{10.57}$$

As far as water, and only in reference to the tubes of the economizer, the value of Re can be considered to lie between 70,000 and 1,000,000; the relative roughness can be assumed to be equal to $0.8\text{–}1.5 \times 10^{-3}$, and the same diagram leads to $\lambda =$ 0.0185–0.0245. Assuming that $\lambda = 0.0215$, we have

$$N_e = 46\zeta \tag{10.58}$$

Clearly, the suggested values of N_e refer only to smoke tubes and coils of superheaters and economizers. Any extrapolation to tubes outside the generator or air and gas ducts (with much greater diameters and ensuing different values of Re and ε) is arbitrary.

10.3 Pressure Drop Through the Tube Banks

When it hits the tube banks (steam-generating bank, superheater, reheater, economizer with smooth tubes, and tube air heater with the gas on the outside), flue gas undergoes a pressure drop that can be computed through the following equation:

$$\Delta p = f_d f_a N \rho \frac{V^2}{2} \tag{10.59}$$

where f_d and f_a stand for two factors that will be discussed later on and N for a number of tube rows crossed by flue gas. With the density ρ in $\mathrm{kg/m^3}$ and the velocity V in m/s, the pressure drop Δp is in Pa. Even in this case it is more convenient to refer to mass velocity G and to density ρ_0 under normal conditions.

Similarly to (10.5) and (10.45), we obtain the following:

$$\Delta p = 1.855 f_d f_a N \frac{G^2}{p\rho_0} \frac{T}{1000}; \tag{10.60}$$

G in kg/m^{2}s, p in bar and the absolute temperature T in K.

As usual, assimilating pressure p to atmospheric pressure we obtain the following through (10.60):

$$\Delta p = 1.83 f_d f_a N \frac{G^2}{\rho_0} \frac{T}{1000}. \tag{10.61}$$

It is equally possible to introduce velocity V_0 under normal conditions. In that case (10.59) leads to

$$\Delta p = 1.855 f_d f_a N \rho_0 \frac{V_0^2}{p} \frac{T}{1000}; \tag{10.62}$$

and assimilating pressure p to atmospheric pressure

$$\Delta p = 1.83 f_d f_a N \rho_0 V_0^2 \frac{T}{1000}. \tag{10.63}$$

The ability to refer to mass velocity or velocity under normal conditions is the very advantageous. Note that along the flue gas pass in the generator temperature is a constant variable which consequently varies the volumetric flow rate of the gas, whereas, of course, the mass flow rate and the volumetric flow rate under normal conditions remain constant. The simple separation of these two flow rates for the pass sections (cross-sectional areas) of the different banks makes it possible to obtain velocity G or velocity V_0 referred to normal conditions.

The different equations above are also valid for the air hitting the tube bank of the air heater. The density under normal conditions is included in (10.58), (10.61), (10.62), and (10.63). For air we have $\rho_0 = 1.293\,\text{kg/Nm}^3$; for flue gas ρ_0 is computed through (7.75) or taken from Table 7.2. The considerations made in Sect. 10.1 in terms of the temperature to consider in (10.50) and (10.63) must be taken into account for steam generating tube banks, as well. In other words, one must consider the mean temperature in the bank instead of the mean temperature between inlet and outlet temperatures.

The latter can be adopted for flue gas flowing through the superheater, the economizer, and the air heater. The same is true for air hitting the tube bank of the air heater.

Factor f_d intervenes only if the number of crossed rows is below10. In fact, for $N \geq 10$ we have $f_d = 1$; for $N < 10$ the values of f_d can be obtained through Fig. 10.24. The arrangement factor f_a is a function of the Reynolds number, of the outside diameter, of the transversal and longitudinal pitch of the tubes, and finally of the type of arrangement.

Let us consider Fig. 10.25 showing an arrangement with inline tubes where the transversal pitch is indicated by s_t and the longitudinal one by s_l. Figure 10.26 helps to obtain f_a for different values of the ratios s_t/d_o and s_l/d_o as a function

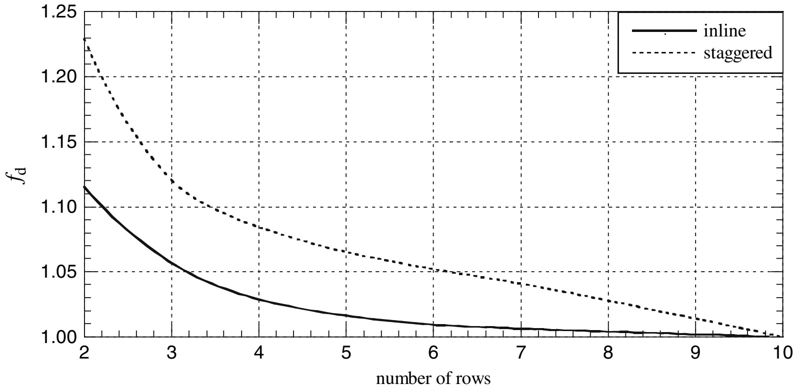
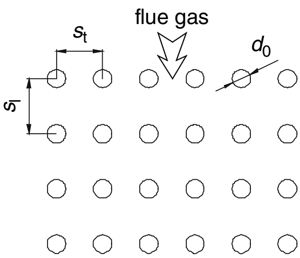


Fig. 10.24 Factor f_d for the pressure drop trough a tube bank

Fig. 10.25 Arrangement with inline tubes



of Re. Figure 10.27 shows a layout with staggered tubes instead, and the diagram in Fig. 10.28 helps to determine the arrangement factor.

Note that except for special situations, s_l/d_o, s_t/d_o, and Re being equal, the values of f_a in staggered arrangements are always higher than those relative to inline tubes. On the other hand, as we know, the staggered arrangement implies higher overall heat transfer coefficients.

The criteria for adopting one over the other arrangement was already discussed in Sect. 8.13 by highlighting the parameters to consider in order to evaluate the problem correctly and to simultaneously take both heat transfer and pressure drops into account. Certain constructive and running requirements notwithstanding, it is a question of adopting the optimal solution for a specific project. In the flue gas passage in a boiler, there are always sudden changes of direction and sometimes even throttling that cause localized pressure drops. Sect. 10.2 provides assistance in defining these drops according to cautionary criteria and by analogy. More specific information cannot be provided, but generally these pressure drops are relatively modest when compared with those deriving from the tube bank. Therefore, it may be enough to conventionally increase the latter by 5–10% in order to forecast the total pressure drop.

In conclusion, even though it is not recommended as far as heat transfer, sometimes the gas flow is parallel to the axis of the tubes and not transversal, as we

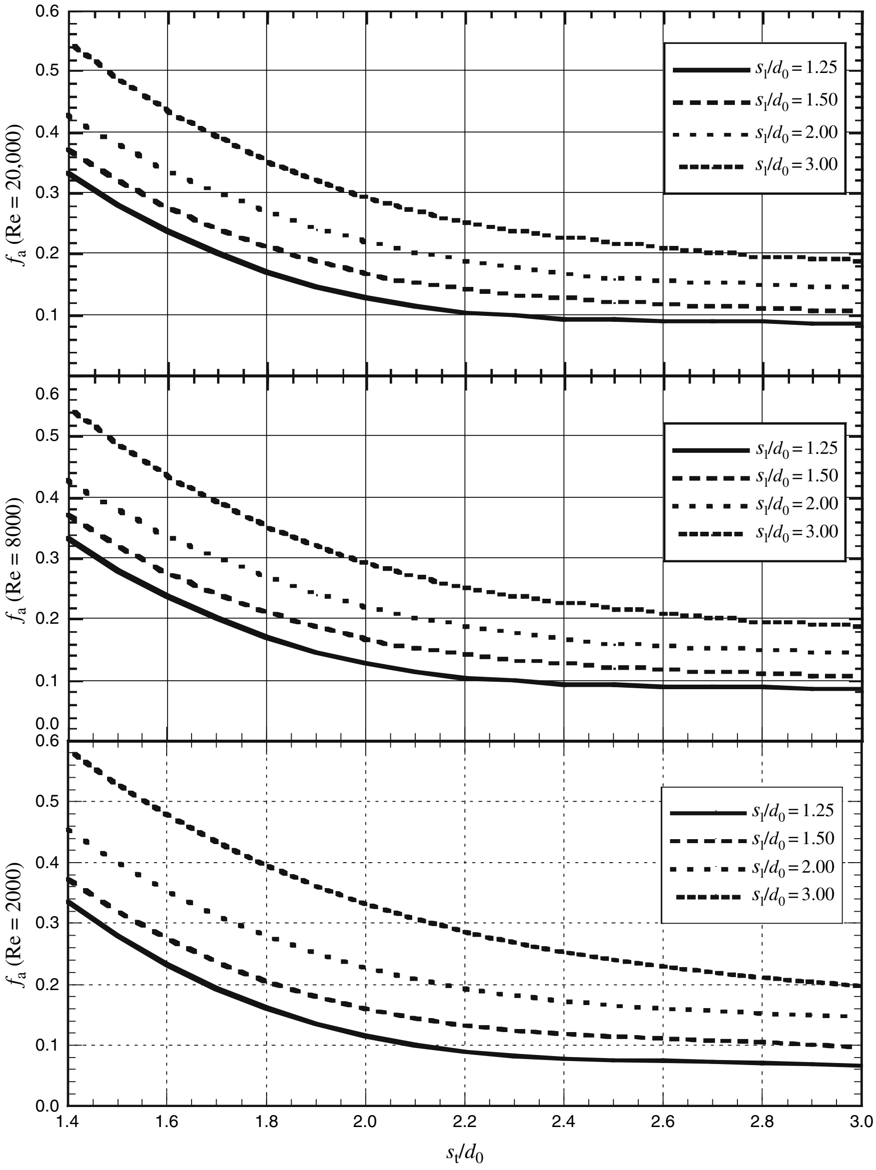


Fig. 10.26 Arrangement factor f_a for inline tubes

assumed so far. In this case we consider the cross-sectional areas shown by the dashed lines in Fig. 10.29 (with staggered arrangement an inline tubes) and the relative wet perimeter (bold line in the figure) and compute the hydraulic diameter through 10.7. The pressure drop is computed based on Sect. 10.1.

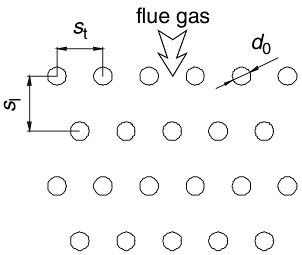


Fig. 10.27 Arrangement with staggered tubes

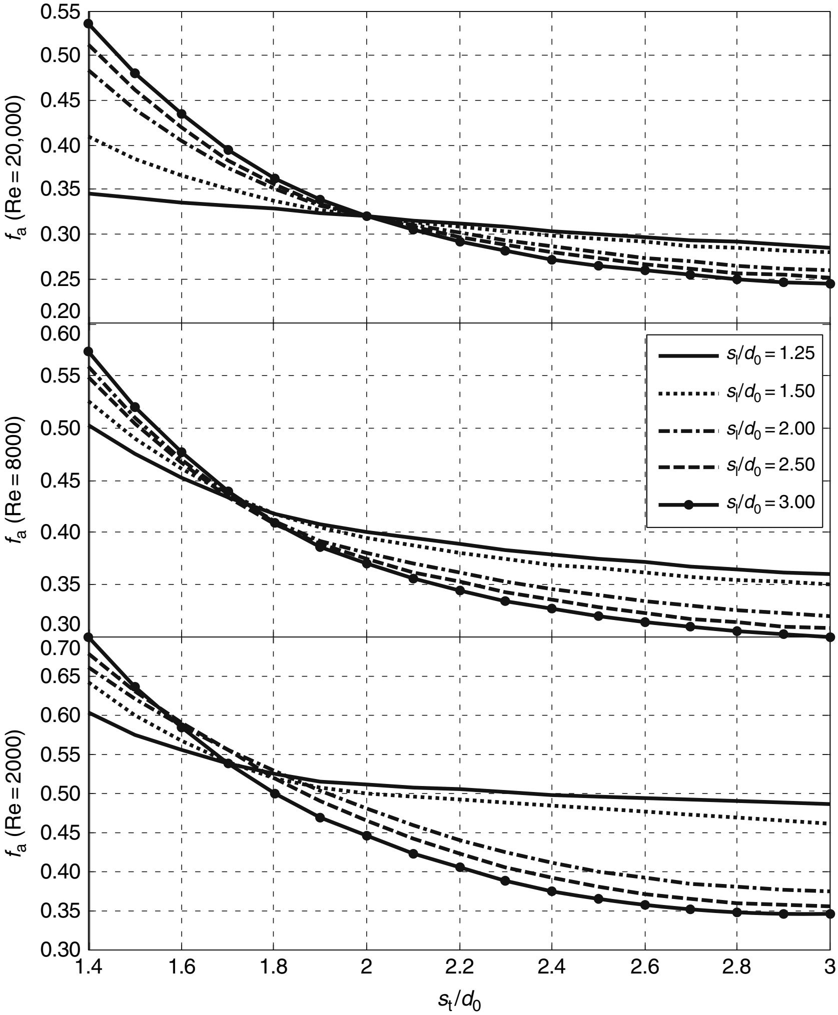
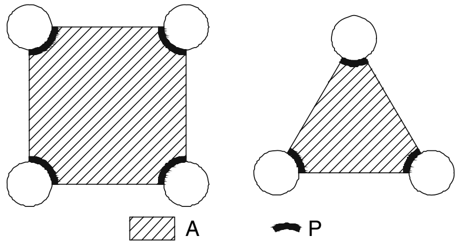


Fig. 10.28 Arrangement factor f_a for staggered tubes

Fig. 10.29



10.4 Pressure Drop in Special Equipments

We saw how to calculate pressure drops in ducts and throughout the generator, at times including an economizer with smooth tubes. This is not sufficient to complete the air and flue gas circuit. Let us look at the circuit of a radiation generator including the installation of all possible equipment in these cases (Figs. 1.6 and 1.7).

First of all, it is required to estimate the pressure drops through the air heater as far as air and flue gas. Given that it belongs to the Ljungstroem type, one is forced to base this on the experimental results of the constructor (generally, the drops amount to about 400–600 Pa for both air and flue gas).

In addition, one must consider the pressure drop due to the burners based on experimental data only. As a reference, note that the pressure difference between the burner box and the furnace required to compensate this drop and to give air the necessary velocity usually ranges from about 800 to 1800 Pa. For burners installed on convection generators, this drop usually ranges from 1200 to 1600 Pa. The pressure drop caused by soot precipitators must be established experimentally and indicated by the constructor of the equipment. Roughly, note that it typically ranges from 400 to 700 Pa.

Finally, even though its computation is quite easy, the pressure drop caused by steam unit heaters is usually indicated by the specialized constructor and mostly is about 300–500 Pa. If it is a convection generator, there could be an economizer with finned tubes after it. In this case one may refer to the general concepts in previous sections taking into account cross-sectional areas of channels (and related throttling) where water and flue gas are flowing. The literature indicates specific calculation criteria able to provide more exact data on pressure drops with finned tubes. This is indispensable during the design phase of an economizer that has not been built yet. On the other hand, if the device has already been experimented, it will be best to refer to results obtained by running plants.

There are no problems in computing pressure drops in tube air heaters. The drops relative to the fluid flowing inside the tubes are computed based on the equations in Sect. 10.1 for distributed drops and based on Sect. 10.2 for inlet and outlet drops in correspondence of the headpieces. The pressure drop relative to the fluid hitting the tube bank is computed based on the criteria described in Sect. 10.3 for tube banks.

If the air heater is a pocket heater, both air and flue gas flow in channels with a rectangular cross-section. Therefore, pressure drops are computed for both circuits

by referring to the hydraulic diameter of the channels and based on the equations in Sect. 10.1 and 10.2 (for the drops at inlet and outlet). As far as pressure drops in the cowlings which reverse the direction of both air and flue gas flowing in the tubes or channels or hit the different cross-sections of the tube bank, orientation is provided by the mass velocity corresponding to the minimum cross-section of the cowling by adopting factor ζ equal to 1.5.

Finally, the last piece of the puzzle is the pressure drop in the chimney. We refer to the equations in Sect. 10.1 for distributed drops considering the roughness of commercial tubes if the chimney is made of steel and the roughness of the masonry duct if the chimney belongs to this category. In addition, one needs to consider the pressure drop at the outlet on the top, according to Sect. 10.2.

10.5 Pumps, Fans, and Chimneys

Feed pumps of steam generators should be sized according to the following criteria. Each generator or group of generators must be fed through two feed devices, one of which is running while the other is a reserve. The two feed devices must not depend on the same energy source unless it is steam operated. Operating both feed devices with electric engines is allowable only when there are at least two independent energy sources and it is possible to rapidly switch from one source to the other. This commutation for steam generators with maximum production greater than 50 t/h must be done automatically. In any case, if the fuel feed devices operated electrically can be switched to a second energy source, it is required that even the main feed device can be switched simultaneously to the same energy source if it is operated electrically.

The flow rate of both main and backup feed pump must be at least equal to the values in Table 10.3. These stand for the percentage of the water flow rate required by the generator or the generators of the group. The latter is equal to the peak output plus the water flow rate of the continuous water drainage and of recirculation on the suction of the pump if they are included.

There are two types of pumps: centrifugal or piston pumps. The former are operated by electric engines or steam turbines, the latter by an electric engine through a reduction gear and adequate crank mechanisms, or by a steam engine. The behavior of centrifugal and piston pumps during runtime is fundamentally different and determines the use of one pump over the other. Within a certain intermediate range both can be used. Note that if a gate valve is installed on the delivery side (as is always the case), the piston pump requires a safety valve to discharge the entire flow rate of the pump.

Figure 10.30 shows the characteristic curves of a centrifugal pump. Note that the head decreases with the increase in flow rate. The efficiency which is rather low at small flow rates increases up to a maximum that usually corresponds to a head just below the maximum. Then it decreases until it reaches zero at maximum flow rate and zero head. The absorbed power grows hand in hand with the increasing flow rate.

Table 10.3 Advised pump flow rate

Maximum steam production of the generator	Pump flow rate as percentage of water flow rate	
	Without automatic regulation	With automatic regulation
Generators with natural or assisted circulation		
till 1 t/h	200%	200%
from 1 t/h to 5 t/h	160%	130%
from 5 t/h to 50 t/h	125%	115%
from 50 t/h to 100 t/h	115%	105%
from 100 t/h to 400 t/h	not allowable	105%
over 400 t/h	not allowable	100%
Generators with forced circulation		
till 1 t/h	not allowable	110%
over 1 t/h	not allowable	100%

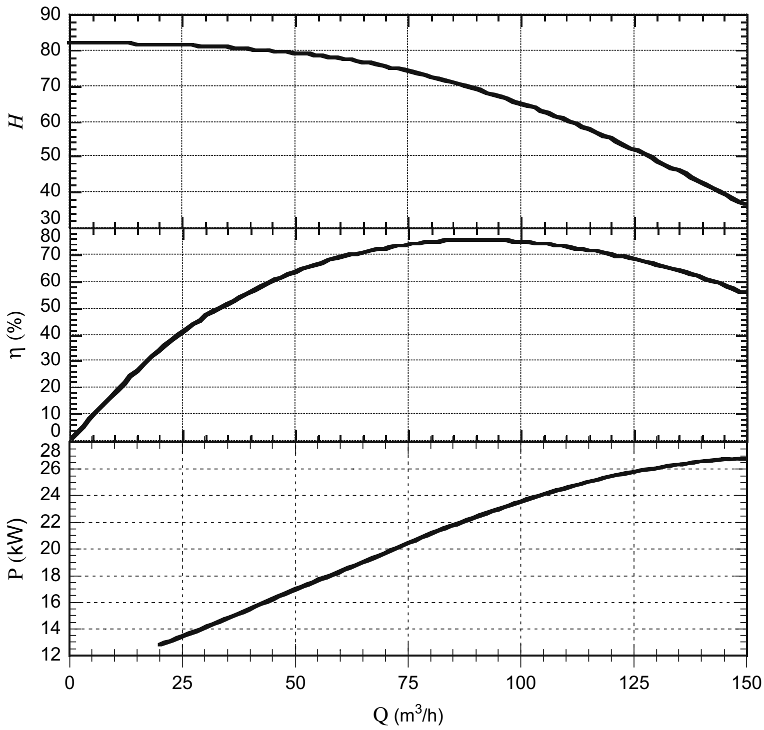


Fig. 10.30 Typical curves of centrifugal pump

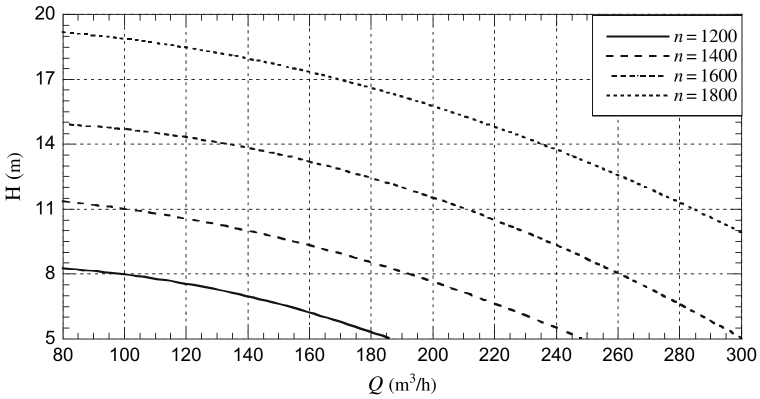


Fig. 10.31 Influence of the number of revolutions on a centrifugal pump

Figure 10.31 shows the impact of the number of revolutions on the head.

There are two types of fans: centrifugal and axial fans. In the former the fluid moving from the inside of the wheel is pushed toward the outside in a radial motion through the blades and discharges into the stator of the fan (Fig. 10.32). In the latter the fluid moves in parallel to the axis of the fan like in a common household fan, except for a collector box where the fluid discharges and generates static pressure (Fig. 10.33).

Centrifugal fans also differentiate themselves from one another by the type of blade. With respect to the radius, the exit angle can be zero, positive, or negative. In

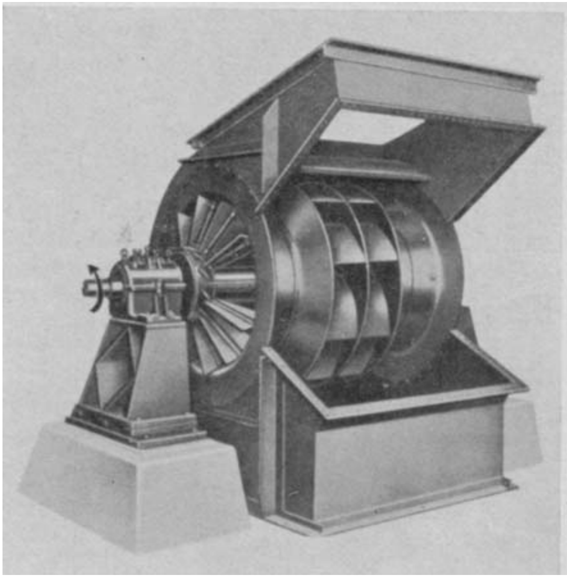


Fig. 10.32 Centrifugal fan (Courtesy of Babcock & Wilcox)

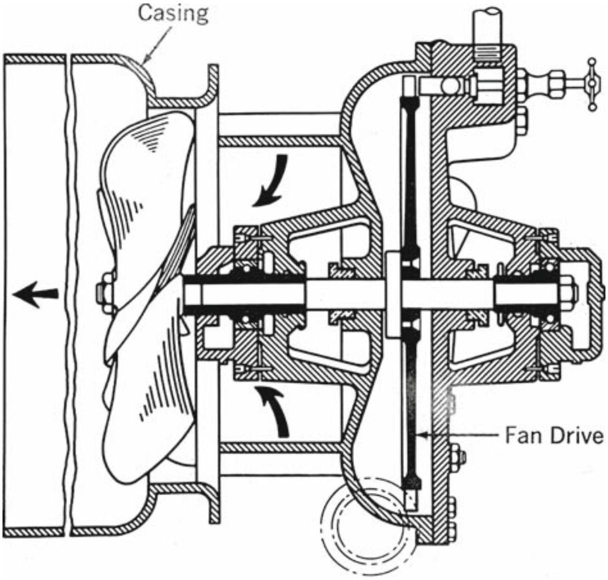


Fig. 10.33 Axial fan (Courtesy of Babcock & Wilcox)

the first case, the blades are radial and can be flat or curved. In the second case, the velocity of the fluid relative to the wheel forms an acute angle with the periphery velocity of the wheel itself. Finally, in the third case, the angle between the two velocities is obtuse (Fig. 10.34).

The first two represent quite typical constructive solutions and the blades of this kind are considered normal. The third solution involves a fan with backward-

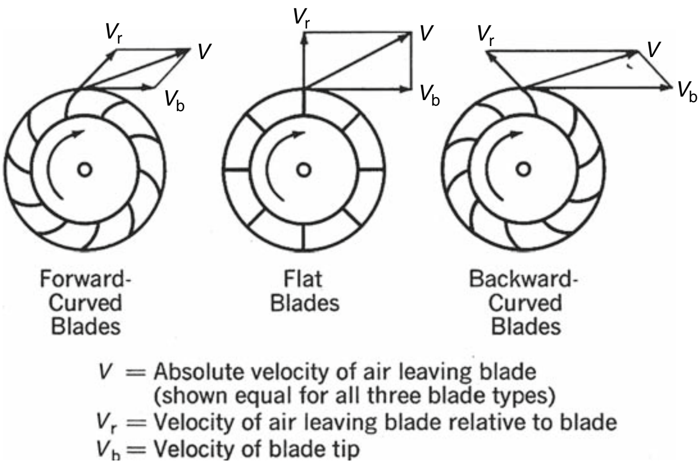


Fig. 10.34 Three general types of centrifugal fans (Courtesy of Babcock & Wilcox)

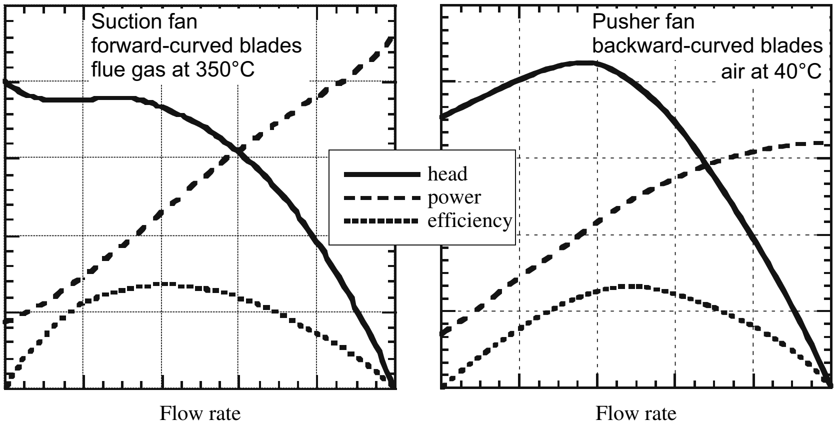


Fig. 10.35 Typical curves of two centrifugal fans

curved blades. Generally, suction fans have normal blades while pusher fans have backward-curved blades.

Figure 10.35 shows the characteristic curves of two fans with the different types of blades.

In the case of forward blades, in the initial part of the curve, the head has an inflection point that may be more or less pronounced or even missing. Regardless, the head is smaller than the one with zero flow rate. The curve representative of the absorbed power is either basically linear (straight) or demonstrates an upward concavity. The maximum efficiency value is generally shifted toward relatively low values of the flow rate.

On the other hand, backward blades imply an increase of the head from zero flow rate up to a value corresponding to the maximum of the curve. Then the head decreases rather rapidly. The absorbed power increase is less compared with the previous solution and shows a clearcut concavity downward. Its characteristics are self-limiting in the sense that it increases slightly with high flow rates. The potential reduction of pressure drop in the circuit and ensuing anomalous increase of the flow rate does not compromise the engine coupled to the fan.

On the other hand, the absorbed power at reduced loads is greater than the one relative to the previous solution. Efficiency reaches its peak in correspondence of a flow rate value that is greater in comparison with the previous case. Based on the characteristics of the circuit (pressure drops as a function of the flow rate), the running condition for the fan is identified by the crossing point of the curve characteristic of the drops with the curve of static pressure of the fan (product of the head by the density of the fluid and gravitational acceleration).

There are two ways to modify the flow rate: by introducing an additional pressure drop or by varying the speed of the fan. The first type of intervention is done through the regulation lock. Greater or lesser opening of the blades changes the pressure drop of the circuit which shifts the crossing point discussed above (Fig. 10.36).

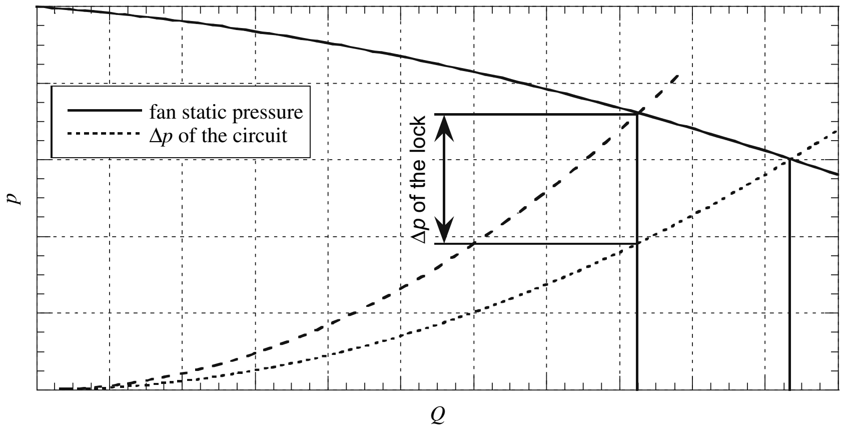


Fig. 10.36 Influence of the air lock

The lock can have rectangular blades and be inserted after the fan in the delivery duct, or it can be round with circle sector blades inserted on the suction (Fig. 10.37). In the latter solution, the dissipated power at reduced loads is less compared with the previous solution.

Flow rate regulation through a lock is the simplest and cheapest solution both in terms of regulation and the coupled engine running at constant speed. It also lends itself to quite simple solutions to set up automatic regulation. In fact, it suffices to

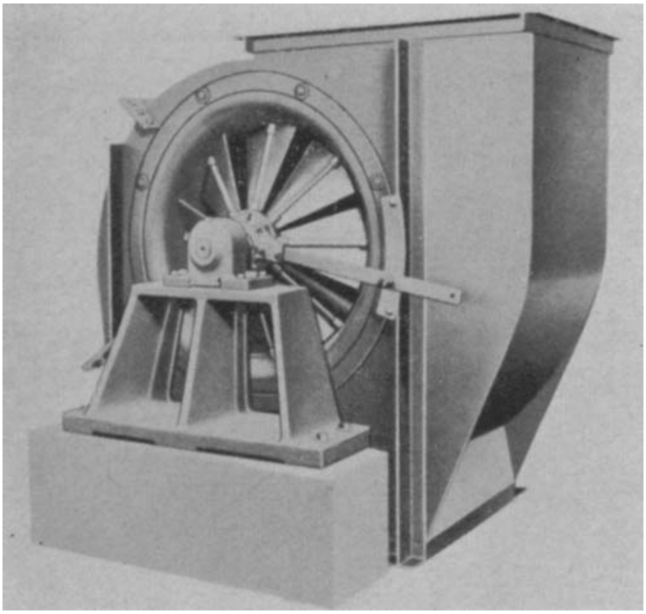


Fig. 10.37 Inlet vanes control output (Courtesy of Babcock & Wilcox)

plan for a servomotor controlled by the regulation plant that makes the blades turn. The lock is the location for dissipated energy because of the increasing pressure drops it introduces into the circuit.

The variation of the flow rate without energy dissipation can be achieved by varying of the fan speed; a lot of energy dissipation is present if the speed variation is not continuous and a regulation lock is still part of the requirements (two-speed engine). Figure 10.38 shows how speed variations modify the characteristic curve of the head (or the curve of static pressure). Thus, the characteristic curve of the circuit crosses the different curves in correspondence of variable flow rates.

Note that, efficiency being equal, variations of the speed imply that the flow rate is proportional to the speed, the head is to the root and consequently the absorbed power is to the cube. Of course, the regulation of the flow rate through speed variations of the fan is quite costly, given that in the simplest case it requires an alternate current engine with two rotation speeds, or the following features in the case of more complex and expensive solutions: magnetic coupling, hydraulic coupling, coupling with a direct current engine or with a variable speed steam turbine.

The fans servicing the generator, besides having high level safety running features because they have to run over long periods of time without interruption, must be sized with a certain leeway both as far as flow rate and head. A safety margin on the flow rate is required both because of the potential necessity to run with more excess air than expected and because of greater than expected air leaks inside the regenerative air heater. In addition, in view of emergency overload, there should be an increased air flow rate. The greater fluid flow rate requires a greater head because of the increasing pressure drops in the circuit. Moreover, their calculation may not be completely close to reality. This means that even the head requested theoretically should be increased. The rule of thumb is to increase the flow rate by at least 10–15% and the head by 15–25%.

Finally, the recommendation is to plan for a higher air temperature at the suction of the pusher fan (it reduces static pressure due to the reduction in air density). An increase by 15 °C should be considered.

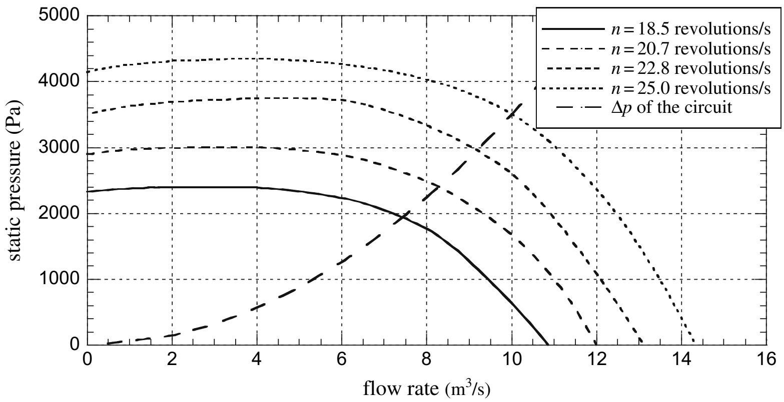


Fig. 10.38 Influence of revolution number on static pressure

The chimney is no longer important as far as suction of flue gas given the considerable pressure drops in the generator. Note that the tendency is to reduce the temperature of the flue gas at the exit of the generator as much as possible. Therefore, the draught of the chimney is modest considering the reduced difference in density between room air and flue gas. The draught p_c of the chimney expressed in Pa is given by:

$$p_c = g\left(\rho_a - \rho_{gm}\right) H_c; \tag{10.64}$$

g stands for the acceleration of gravity in m/s^2, ρ_a and ρ_{gm} for the air density and the mean density of the flue gas in kg/m^3, and H_c for the height of the chimney in m.

Considering the air at $20\,°C$, so that $\rho_a = 1.2\,kg/m^3$, and adopting a value of the density of flue gas under normal conditions equal to $1.3\,kg/Nm^3$, considering that the pressure is practically equal to atmospheric pressure, we have

$$\rho_{gm} = 1.3\,\frac{273}{t_{gm}+273}. \tag{10.65}$$

From (10.64) and with $g = 9.807\,m/s^2$ we have

$$p_c = \left(11.77 - \frac{3480}{t_{gm}+273}\right) H_c. \tag{10.66}$$

With a temperature of the flue gas equal, for instance, to $180\,°C$ the draught is equal to 4.1 Pa/m. This is clearly a modest value. Thus, the draught is quite modest even for very high chimneys in comparison to the pressure drops between furnace and chimney amounting to a several thousand Pa in big generators and to about 1000 Pa in small power ones. At any rate, the chimney has its own distributed pressure drop and exit drop at the top that reduce its useful draught. Therefore, the main purpose of the chimney is to direct the flue gas to a certain height.

It can be made of metal sheet for small units with a height ranging from 20 to 40 m. In thermoelectric plants, it is made of concrete, coated inside with refractory material, and sometimes equipped with a hollow space between the latter and the shaft made of steel concrete used for the passage of cooling air. The goal of the latter is to reduce the entity of thermal stresses in the shaft. There are also chimneys with a shaft in steel concrete and one or more cylindrical elements in metal sheet supported by the shaft, yet completely independent from it. The height of these chimneys is always considerable for ecological reasons, particularly if environmental conditions are unfavorable to dispersion of polluting substances. Some of them are 200 m tall and beyond.

Finally, note that the base of metallic chimneys can be done in such a way to contain a wheel and potentially even a regulation lock. In that case, we speak of a mechanical chimney. The simple placement of the cylinder over the base creates both the suction fan and the chimney, once the belt pulley is connected to an electric engine.

10.6 Natural Circulation

Both existing and new convection generators have natural circulation. The same is true for many existing radiation generators. Several of the new ones have natural circulation, as well. Considering that the basic principles of assisted circulation coincide with those of natural circulation except for the circulation pump in the circuits of water and saturated steam, the importance of natural circulation is quite understandable. Therefore, let us take a look at its physics laws and leave the definition of the conditions for satisfactory natural circulation to occur to the end.

First of all, we examine the simple circuit in Fig. 10.39 where the downcomers outside the flue gas pass and where warm water is passing through feed the lower header. The steam-generating tubes (also called raisers) under radiation from the flame start from here and redirect the water and steam mix forming inside them directly into the drum.

H stands for the head between the water surface in the drum and the axis of the lower header, ρ_{w} for water density, ρ_{mm} for the mean density of the water–steam mix, Δp_{d} for the pressure drop in the downcomers, and Δp_{r} for the pressure drop in the raisers.

With reference to the water in the downcomers, the hydrostatic pressure in correspondence of the header is equal to

$$p_{\mathrm{d}} = \rho_{\mathrm{w}} g H \tag{10.67}$$

where g stands for the acceleration of gravity assumed to be $9.807\,\mathrm{m/s^2}$.

Similarly, as far as the raisers, the corresponding hydrostatic pressure is equal to

$$p_{\mathrm{r}} = \rho_{\mathrm{mm}} g H. \tag{10.68}$$

The two pressures p_{d} and p_{r} are not identical, and therefore, the circuit is not balanced. This generates an anticlockwise circulation, given that $p_{\mathrm{r}} < p_{\mathrm{d}}$ because the density of the mix is less than that of water due to the presence of steam.

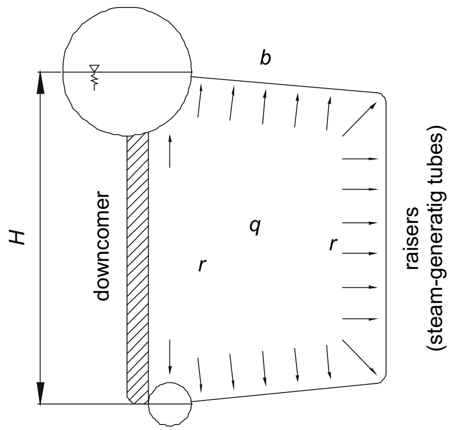


Fig. 10.39 Elementary circuit by natural circulation

Balance is reestablished when the difference between p_d and p_r is compensated by the pressure drops generated in the circuit by circulation. In other words, recalling (10.67) and (10.68) and the significance of Δp_d and Δp_r,

$$g\,(\rho_w - \rho_{mm})\,H = \Delta p_d + \Delta p_r \tag{10.69}$$

A given amount of heat transferred to the tubes generates a given amount of steam. Density ρ_{mm} is a function of the steam output and of the water flow rate feeding them through the downcomers. It goes up with the increase of the ratio between water entering the tubes and the steam output, called circulation ratio. Therefore, the term on the left hand side of the equal sign decreases with an increase in circulation ratio.

On the other hand, the pressure drops increase with the water flow rate or the water–steam mix flow rate in the downcomers and in the raisers, in other words with the circulation ratio. This determines a balance condition that can be identified through the following computation.

We indicate the heat transferred within the time unit to the raisers with q. If r stands for the vaporization heat at the running pressure set in advance, the mass flow rate of the steam at the outlet of the raisers M_{se} is given by

$$M_{se} = \frac{q}{r}, \tag{10.70}$$

where q in expressed in kW, r in kJ/kg, and M_{se} in kg/s.

If M_m stands for the mass flow rate of the steam–water mix in the raisers (of course, equal to the mass flow rate of water in the downcomers M_w) and R for the circulation ratio, the latter is given by

$$R = \frac{M_m}{M_{se}}. \tag{10.71}$$

We introduce two additional quantities, the percentage by mass and the percentage by volume of steam in the mix at the outlet of the raisers.

By indicating the former with π_m we have

$$\pi_m = \frac{M_{se}}{M_m} \times 100 = \frac{100}{R} (\text{in } \%); \tag{10.72}$$

π_v stands for the latter and v_w and v_s for the specific volume of water and steam, respectively.

The volumetric mass flow Q_{se} of the steam at the outlet of the raisers is given by

$$Q_{se} = M_{se} v_s. \tag{10.73}$$

The mass flow rate M_{we} of the water contained in the mix is equal to

$$M_{we} = M_m - M_{se}, \tag{10.74}$$

whereas the volumetric one Q_{we} is equal to

$$Q_{\mathrm{we}} = (M_{\mathrm{m}} - M_{\mathrm{se}})\, v_{\mathrm{w}}. \tag{10.75}$$

Then, in view of the significance of π_{v}, we have

$$\pi_{\mathrm{v}} = \frac{Q_{\mathrm{se}}}{Q_{\mathrm{we}} + Q_{\mathrm{se}}} \times 100 = \frac{M_{\mathrm{se}} v_{\mathrm{s}}}{(M_{\mathrm{m}} - M_{\mathrm{se}})\, v_{\mathrm{w}} + M_{\mathrm{se}} v_{\mathrm{s}}} \times 100. \tag{10.76}$$

After a series of steps that are not shown here, we obtain

$$\pi_{\mathrm{v}} = \frac{100}{\left(\frac{M_{\mathrm{m}}}{M_{\mathrm{se}}} - 1\right) \frac{v_{\mathrm{w}}}{v_{\mathrm{s}}} + 1}. \tag{10.77}$$

Finally, recalling (10.72), we have

$$\pi_{\mathrm{v}} = \frac{100}{\left(\frac{100}{\pi_{\mathrm{m}}} - 1\right) \frac{v_{\mathrm{w}}}{v_{\mathrm{s}}} + 1} (\text{in } \%). \tag{10.78}$$

By analogy, we may write that

$$\pi_{\mathrm{m}} = \frac{100}{\left(\frac{100}{\pi_{\mathrm{v}}} - 1\right) \frac{v_{\mathrm{s}}}{v_{\mathrm{w}}} + 1} (\text{in } \%). \tag{10.79}$$

If ρ_{s} stands for the steam density, the mass flow rate M_{m} of the mix is given by

$$M_{\mathrm{m}} = Q_{\mathrm{we}} \rho_{\mathrm{w}} + Q_{\mathrm{se}} \rho_{\mathrm{s}}. \tag{10.80}$$

On the other hand, the volumetric flow rate Q_{me} of the mix at the outlet of the raisers is equal to

$$Q_{\mathrm{me}} = Q_{\mathrm{we}} + Q_{\mathrm{se}}. \tag{10.81}$$

Then, the density of the mix at the outlet of the raisers is given by

$$\rho_{\mathrm{me}} = \frac{Q_{\mathrm{we}} \rho_{\mathrm{w}} + Q_{\mathrm{se}} \rho_{\mathrm{s}}}{Q_{\mathrm{we}} + Q_{\mathrm{se}}}, \tag{10.82}$$

which can be written as follows:

$$\rho_{\mathrm{me}} = \frac{Q_{\mathrm{se}}}{Q_{\mathrm{we}} + Q_{\mathrm{se}}} \left(1 + \frac{Q_{\mathrm{we}} \rho_{\mathrm{w}}}{Q_{\mathrm{se}} \rho_{\mathrm{s}}}\right) \rho_{\mathrm{s}}. \tag{10.83}$$

In other words,

$$\rho_{\mathrm{me}} = \frac{Q_{\mathrm{se}}}{Q_{\mathrm{we}} + Q_{\mathrm{se}}} \left(1 + \frac{M_{\mathrm{we}}}{M_{\mathrm{se}}}\right) \rho_{\mathrm{s}}, \tag{10.84}$$

or

$$\rho_{\mathrm{me}} = \frac{Q_{\mathrm{se}} 100}{Q_{\mathrm{we}} + Q_{\mathrm{se}}} \frac{M_{\mathrm{we}} + M_{\mathrm{se}}}{M_{\mathrm{se}} 100} \rho_{\mathrm{s}}. \tag{10.85}$$

Recalling (10.72), (10.74) and (10.76), we finally have:

$$\rho_{\mathrm{me}} = \frac{\pi_{\mathrm{v}}}{\pi_{\mathrm{m}}} \rho_{\mathrm{s}}. \tag{10.86}$$

Equation (10.86) helps to compute the density of the mix at the outlet of the raisers when π_{m} and π_{v} are known (of course, ρ_{s} is known). The value of π_{m} is quickly calculated as a function of the preset circulation ratio (as we shall see later on this is indispensable to design and verification calculations) based on (10.72). The value of π_{v} is computed through (10.78) once π_{m} is known.

Knowing the value of ρ_{me}, the mean density of the water–steam mix ρ_{mm} included in (10.69) is assumed to be equal to

$$\rho_{\mathrm{mm}} = \frac{\rho_{\mathrm{w}} + \rho_{\mathrm{me}}}{2}, \tag{10.87}$$

given that at the inlet of the raisers the density of the mix matches the density of the water.

Now we introduce factor τ equal to the ratio between the mean volumetric flow rate of the steam and the mean volumetric flow rate of the mix. If Q_{sm} and Q_{mm} stand for them, respectively, by definition τ is given by

$$\tau = \frac{Q_{\mathrm{sm}}}{Q_{\mathrm{mm}}}. \tag{10.88}$$

The mean density of the steam–water mix can also be defined as the ratio between the mass flow rate M_{m} and the mean volumetric flow rate, that is,

$$\rho_{\mathrm{mm}} = \frac{M_{\mathrm{m}}}{Q_{\mathrm{mm}}}. \tag{10.89}$$

On the other hand, the mean volumetric flow rate of the water in the mix Q_{wm} is given by

$$Q_{\mathrm{wm}} = Q_{\mathrm{mm}} - Q_{\mathrm{sm}}. \tag{10.90}$$

The mass flow rate can be expressed as follows:

$$M_{\mathrm{m}} = Q_{\mathrm{sm}}\rho_{\mathrm{s}} + Q_{\mathrm{wm}}\rho_{\mathrm{w}} = Q_{\mathrm{sm}}\rho_{\mathrm{s}} + (Q_{\mathrm{mm}} - Q_{\mathrm{sm}})\,\rho_{\mathrm{w}}. \tag{10.91}$$

Thus, (10.89) can be written as follows:

$$\rho_{\mathrm{mm}} = \frac{Q_{\mathrm{sm}}\rho_{\mathrm{s}} + (Q_{\mathrm{mm}} - Q_{\mathrm{sm}})\,\rho_{\mathrm{w}}}{Q_{\mathrm{mm}}}. \tag{10.92}$$

Recalling (10.88) we obtain

$$\rho_{\mathrm{mm}} = \tau\rho_{\mathrm{s}} + (1-\tau)\,\rho_{\mathrm{w}}. \tag{10.93}$$

From (10.93) through obvious steps we obtain

$$\tau = \frac{\rho_w - \rho_{mm}}{\rho_w - \rho_s}. \tag{10.94}$$

After calculating ρ_{mm} with (10.87) it is possible to compute τ.

Factor τ is interesting in terms of the computation of the mean viscosity of the water–steam mix. In fact, this is computed as the weighted average of the viscosity of water and steam referred to the mean volumetric flow rates of both fluids. μ_w, μ_s, and μ_{mm} stand for the viscosity of water, steam, and steam–water mix average, respectively.

Note that in (10.93) even the mean density of the mix is given by the weighted average of the density of water and steam referred, of course, to the mean volumetric flow rates of the components.

Similarly, we have

$$\mu_{mm} = \tau\mu_s + (1-\tau)\,\mu_w. \tag{10.95}$$

We indicate d_i as the inside diameter of the raisers and G_r as the mass velocity. If A_r stands for the global cross-sectional area of the raisers considered in the calculation, and recalling the meaning of M_m, mass velocity is given by

$$G_r = \frac{M_m}{A_r}. \tag{10.96}$$

The Reynolds number relative to the raisers is therefore given by

$$\mathrm{Re} = \frac{G_r d_i}{\mu_{mm}} \tag{10.97}$$

with G_r in kg/m^{2}s, d_i in m, and μ_{mm} in kg/ms.

The values of μ_w and μ_s which influence the value of μ_{mm} can be obtained, for instance, from the publication "Properties of water and steam in SI-units." After computing the Reynolds number and obtaining the value of the relative roughness ε through the diagram in Fig. 10.1 or through (10.21), it is possible to compute the distributed pressure drop according to Sect. 10.1. To that extent, it is quite convenient to use (10.3) because it includes mass velocity. Of course, ρ_{mm} is introduced as the value of the density.

Then one must calculate the inlet loss in the raisers by applying the criteria discussed in Sect. 10.2. Even in this case, it will be best to refer to (10.44) and to consider the density of the water. Moreover, one needs to compute the outlet pressure drops. Density ρ_{me} relative to the outlet of the raisers must be added to (10.44). As far as the values of ζ, in both cases caution is advisable by adopting $\zeta = 0.5$ for the inlet and $\zeta = 1$ for the outlet.

Finally, one computes the drops in the curves. The criteria in Sect. 10.2 are still true and the mean value ρ_{mm} as density can be used for all curves. We see that for the computation of the pressure drops based on constant mass velocity, it suffices to consider the different values of the density in the equations. If Δp_{rd} stands for

the distributed drops and Δp_{ri}, Δp_{ro}, and Δp_{rc} stand for concentrated drops at inlet, outlet, and the curves, respectively, the total pressure drop Δp_{r} in the raisers is equal to

$$\Delta p_{\mathrm{r}} = \Delta p_{\mathrm{rd}} + \Delta p_{\mathrm{ri}} + \Delta p_{\mathrm{ro}} + \Delta p_{\mathrm{rc}}. \tag{10.98}$$

Note that if there are deflecting plates, cyclones, and so on at the outlet of the tubes in the drum, the relative pressure drops must be considered, too. These can be implied from experimental data. The pressure drops in the downcomers are computed in a similar way. The calculation is made easier by the fact that density is constant (ρ_{w}).

Based on the total cross-sectional area of the water passage A_{d}, one computes mass velocity given by

$$G_{\mathrm{d}} = \frac{M_{\mathrm{w}}}{A_{\mathrm{d}}} = \frac{M_{\mathrm{m}}}{A_{\mathrm{d}}}. \tag{10.99}$$

The distributed and concentrated drops relative again to inlet, outlet, and the curves are computed using the usual equations in Sect. 10.1 and 10.2.

Similarly, if the distributed drops are indicated as Δp_{dd} and the drops relative to inlet, outlet, and the curves as Δp_{di}, Δp_{do}, and Δp_{dc}, respectively, the total pressure drop is given by

$$\Delta p_{\mathrm{d}} = \Delta p_{\mathrm{dd}} + \Delta p_{\mathrm{di}} + \Delta p_{\mathrm{do}} + \Delta p_{\mathrm{dc}}. \tag{10.100}$$

This process leads to the values of ρ_{mm}, Δp_{d}, and Δp_{r} included in (10.69).

Once the running pressure, thus the values of r, ρ_{w}, and ρ_{s} (and consequently v_{w} and v_{s}), is set and the value of the heat q transferred to the raisers is defined, the three quantities above are only a function of the circulation ratio. We already pointed out that the term on the left hand side of (10.69) decreases with the increase of R, whereas the term on the right hand side goes up. The qualitative behavior of both terms as a function of R is therefore the one included in (10.40); the crossing point of the curves identifies the balance condition matched by the characteristic value of R in the circuit.

To do the verification calculation of the circuit, one must select the different values of R and execute the computation of ρ_{mm}, Δp_{d}, and Δp_{r} for each one. This allows to trace the two curves in Fig. 10.40 and to identify the value of R relative to the circuit in question. At this point one must decide whether the resulting value is sufficient to guarantee correct functioning of the generator, that is, to rule out anomalous superheating of the tubes caused by insufficient cooling by the water–steam mix. The recommendation is to not exceed the values of π_{v} shown in the diagram of Fig. 10.41 as a function of absolute pressure. Based on (10.79), it is possible to compute π_{m} once π_{v} and running pressure (thus, v_{w} and v_{s}) are known. The value of R is computed through (10.72) from π_{m}.

Following this process, we computed R by including its values in the diagram of Fig. 10.42. They stand for minimum values. The design computation of the circulation is done as follows. The desired value of R is set based on the diagram of

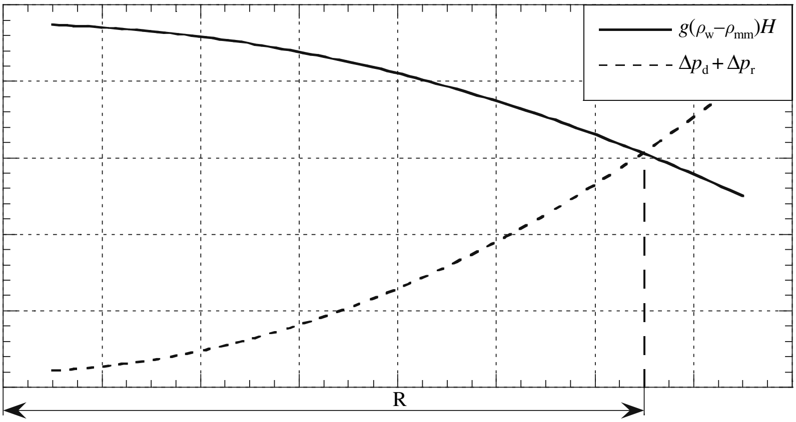


Fig. 10.40

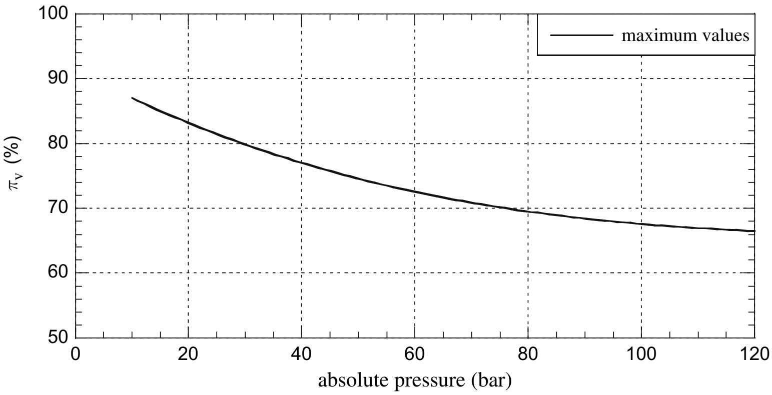


Fig. 10.41 Maximum values of π_v

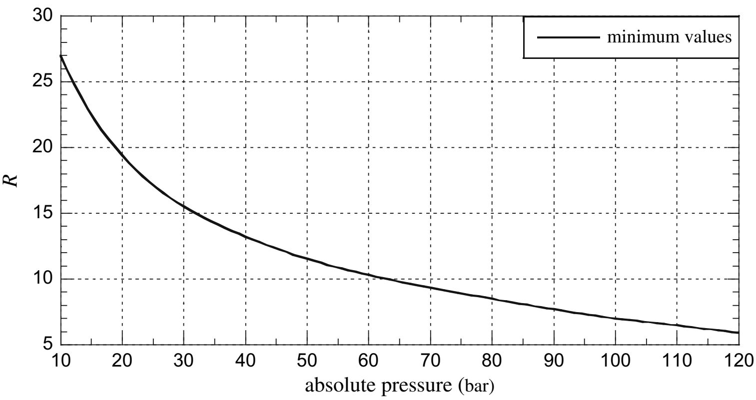


Fig. 10.42 Minimum values of R

Fig. 10.42, and ρ_{mm} and Δp_r are calculated based on this value. Based on (10.69), to ensure that the value of R is greater or equal to the set value, we must have

$$\Delta p_d \leq g\,(\rho_w - \rho_{mm})\,H - \Delta p_r. \qquad (10.101)$$

It is a question of sizing the downcomers in such a way to agree with (10.101).

Note here that the global cross-sectional areas being equal, few tubes of large diameter cause a smaller pressure drop compared to many tubes of small diameter. In fact, recall (10.1) where d_i is in the denominator; as far as λ, an increase in diameter increases Re and reduces ε; Figure 10.2 demonstrates that even λ decreases. Of course, the logical tendency to plan for few tubes with a large diameter should not exasperated because it is very important to distribute the water in the raisers as evenly as possible.

Sometimes, it is preferable to have a sufficient number of downcomers in that respect. All depends on the constructive characteristics of the generator.

Note that (10.69) can be written as follows:

$$(\rho_w g H - \Delta p_d) + (-\rho_{mm} g H - \Delta p_r) = 0. \qquad (10.102)$$

If we consider the difference in geodetic height between the inlet and the outlet of the fluid in the downcomers and in the raisers, given its direction, we see that the difference in height Δz_d relative to the downcomers is equal to H, whereas the one relative to the raisers (Δz_r) is equal to $-H$. Therefore, we can write (10.102) as follows:

$$(\rho_w g \Delta z_d - \Delta p_d) + (\rho_{mm} g \Delta z_r - \Delta p_r) = 0. \qquad (10.103)$$

This highlights a characteristic factor of each circuit branch with the dimensions of a pressure that we call P. It is generically expressed as follows:

$$P = \rho g \Delta z - \Delta p \qquad (10.104)$$

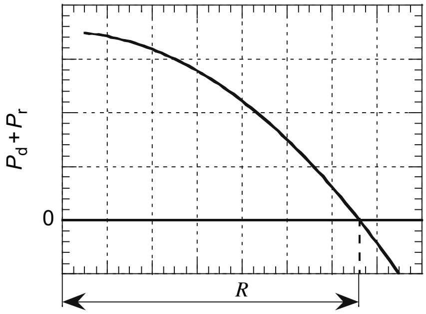
where ρ stands for the density characteristic of the branch in question, equal to the constant density of the fluid in the case of water and to the mean density in the case of a water–steam mix.

If P_d and P_r stand for the values of this factor for downcomers and raisers, (10.67) is synthesized as follows:

$$P_d + P_r = 0. \qquad (10.105)$$

It is easy to establish that given zero velocity on the water surface in the drum and at the outlet of downcomers and raisers (considering that the entire kinetic energy is lost at the outlet), P_d represents the difference in pressure between the axis of the lower collector and the drum (positive) caused by the hydrostatic pressure of the water column in the downcomers minus the pressure drops. P_r represents the difference in pressure (negative) between the drum and the axis of the collector (inlet of the raisers) caused by hydrostatic pressure of the water–steam mix column minus the pressure drops.

Fig. 10.43



Therefore, (10.105) is completely justified. If we diagram the left hand term in (10.105) as a function of R, we obtain a curve shown in Fig. 10.43 with its qualitative pattern. The value of R of the circuit in question is identified by the intersection point with the axis of the abscissa.

Moreover, factor P is not only characteristic of a branch in its entirety but also of any single section of the branch (it no longer indicates the difference in pressure between extremities because of the influence of velocity).

Let us consider the generic branch shown in Fig. 10.44 characterized by a broken line. It can stand for a downcomer or a group of downcomers sharing the same geometric characteristics, or a group of identical raisers, or one or more return tubes (as we shall see later on).

Considering the generic section of the branch, we can compute its characteristic factor. With reference to (10.104), density ρ is the mean one of the fluid in the section, Δz is the difference in geodetic height between the extremity of section where the fluid enters and the extremity where the fluid exits, Δp is equal to the distributed pressure drop along the section plus the drops concentrated at the extremities, such as an inlet, outlet, or curve. Of course, concentrated pressure drops must be counted only once. In other words, if they are counted on both ends for a section, in the previous and next sections, the drops at the outlet and at the inlet, respectively, will not be counted.

After computing the factors P for all sections of the branch that we generically indicate with t, the characteristic factor of the branch is given by

$$P_t = \sum_t P \tag{10.106}$$

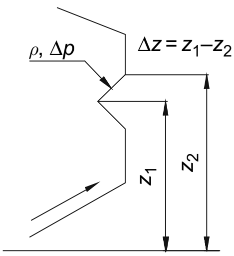


Fig. 10.44

where factor P stands for the generic section; the summation is extended to all sections of branch t.

For instance, if d and r are the branches corresponding to the downcomers and the raisers, respectively, based on (10.105) the condition of balance is represented by the following equation:

$$\sum_d P + \sum_r P = 0. \tag{10.107}$$

The calculation done through the factors P, characteristic of any section of the circuit, makes it possible, first of all, to compute circulation more accurately. Specifically, as far as the downcomers (and the return tubes, as we shall see), there is no difference between summation $\sum_d P$ and the value of factor P computed by including the entire difference in height of the circuit and the global pressure drops. In other words, recalling the significance of Δz_d and Δp_d and (10.103):

$$\sum_d P = \rho_w g \Delta z_d - \Delta p_d. \tag{10.108}$$

In fact, ρ_{w} is constant and this means that the summation of the generic terms $\rho\Delta z$ of any section corresponds to $\rho_{\mathrm{w}}\Delta z_d$. The summation of the different distributed and concentrated drops of any section necessarily corresponds to Δp_d, too. This is not true for the raisers. Previously, the computation was done considering a mean value of density (ρ_{mm}) computed as the average between the density at the inlet and outlet of the entire branch. Moreover, the pressure drops on the curves have also been computed with reference to ρ_{mm}.

But the summation of the $\rho\Delta z$ products relative to every section is not equal to the product of the mean density (not even if the exact mean value along the tube were to be computed instead of the average between extreme values) multiplied by the entire geodetic height difference of the branch. In other words,

$$\sum_r \rho\Delta z \neq \rho_{\mathrm{mm}}\Delta z_r. \tag{10.109}$$

In fact, the influence on the value of the summation of the vertical sections (with a high value of Δz) and of the less tilted sections with respect to the horizontal line is quite different. In other words, the values of ρ that count are those in the vertical sections. The value of the density that should be multiplied with Δz_r to obtain the same value of the summation is a weighted mean value that should take the geometry of the different sections into account (as in more or less tilt).

As far as pressure drops, the one at inlet and at outlet are correct. The distributed ones are equal to the summation of the various sections only if ρ_{mm} is the actual mean value, not the average between extreme values. Note that the density of the water–steam mix, especially if the circulation ratio is low, undergoes a sudden reduction at the beginning of the steam production, while the density reduction is less accentuated after that. In other words, the behavior of ρ along the branch is nothing but linear. Therefore, it is not correct to adopt the average between extremes as the mean value.

Then, as far as drops in the curves, an exact calculation is possible only based on the local density of the mix. Thus, with reference to (10.103) and (10.107):

$$\sum_r P \neq \rho_{\mathrm{mm}} g \Delta z_r - \Delta p_r. \tag{10.110}$$

Recalling that (10.101) reflects (10.69), we establish that the calculation process described at the beginning of the section is gross and is admissible only by initial approximation or for general orientation and exclusively for elementary circuits. It can absolutely not be considered for circuits with parallel braches, as we shall see later on.

The density of the water–steam mix in any point of the branch can be computed as follows.

We indicate β as the ratio between the mass flow rate of the steam M_{s} in that spot and the mass flow rate of the steam M_{se} at the outlet of the branch, that is,

$$\beta = \frac{M_{\mathrm{s}}}{M_{\mathrm{se}}}. \tag{10.111}$$

If M_{m} is the mass flow rate of the water-steam mix, we have:

$$M_{\mathrm{s}} = \beta M_{\mathrm{se}}; \tag{10.112}$$

$$M_{\mathrm{w}} = M_{\mathrm{m}} - M_{\mathrm{s}} = M_{\mathrm{m}} - \beta M_{\mathrm{se}}; \tag{10.113}$$

M_{w} stands for the mass flow rate of the water.

The specific volume v^* of the mix is equal to:

$$v^* = \frac{M_{\mathrm{s}} v_{\mathrm{s}} + M_{\mathrm{w}} v_{\mathrm{w}}}{M_{\mathrm{m}}} \tag{10.114}$$

where v_{s} and v_{w} are, usual, the specific volumes of the steam and the water.

Based on (10.112) and (10.113) we have

$$v^* = \beta \frac{M_{\mathrm{se}}}{M_{\mathrm{m}}} v_{\mathrm{s}} + \left(1 - \beta \frac{M_{\mathrm{se}}}{M_{\mathrm{m}}}\right) v_{\mathrm{w}}. \tag{10.115}$$

Recalling (10.71), we obtain the following from (10.115):

$$v^* = \frac{1}{R}\left[\beta v_{\mathrm{s}} + (R - \beta)\, v_{\mathrm{w}}\right]. \tag{10.116}$$

Finally, the density of the water-steam mix ρ^* is given by:

$$\rho^* = \frac{R}{\beta v_{\mathrm{s}} + (R - \beta)\, v_{\mathrm{w}}}. \tag{10.117}$$

The diagram in Fig. 10.45 shows the values of ρ^* for $p = 50$ bar and for various values of R as a function of β. We determine what was said earlier, that is, that the behavior of the density is not at all linear, especially for low values of R. Let us

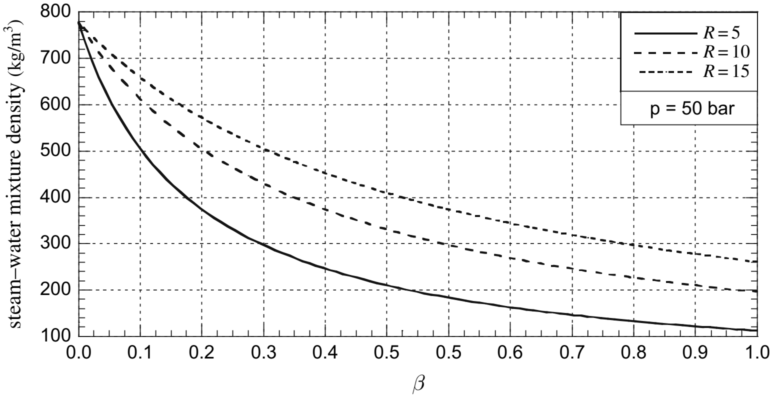


Fig. 10.45

now examine a section. β_1 stands for the value of β relative to the mass flow rate of steam entering the section, and β_2 stands for the value of β relative to the mass flow rate of steam exiting the section.

The mean density of the water–steam mix in the section simply indicated by ρ is equal to:

$$\rho = \frac{1}{\beta_2 - \beta_1} \int_{\beta_1}^{\beta_2} \rho^* d\beta. \tag{10.118}$$

Resolving the integral we obtain:

$$\rho = \frac{R}{(\beta_2 - \beta_1)(v_s - v_w)} \left| \log_e \left[\beta v_s + (R - \beta) v_w \right] \right|_{\beta_1}^{\beta_2}. \tag{10.119}$$

Therefore,

$$\rho = \frac{R}{(\beta_2 - \beta_1)(v_s - v_w)} \log_e \frac{\beta_2 v_s + (R - \beta_2) v_w}{\beta_1 v_s + (R - \beta_1) v_w}. \tag{10.120}$$

The value of ρ computed through (10.120) must be introduced in (10.104) to obtain the value of factor P relative to the section in question.

Equation (10.120) can also be written as follows, thus obtaining the equation suggested by Ledinegg:

$$\rho = \rho_w \frac{\frac{R}{\beta_2}}{\left(1 - \frac{\beta_1}{\beta_2}\right)\left(\frac{\rho_w}{\rho_s} - 1\right)} \log_e \frac{\frac{R}{\beta_2} + \left(\frac{\rho_w}{\rho_s} - 1\right)}{\frac{R}{\beta_2} + \frac{\beta_1}{\beta_2}\left(\frac{\rho_w}{\rho_s} - 1\right)}. \tag{10.121}$$

Now we take a look at the details of how to proceed. We consider the different sections of the branch and we estimate the transferred heat q. Note that it may also correspond to a thermal flux that varies from section to section. Even from this point

of view the calculation by section is much more sophisticated and makes it possible to highlight potential intensity variations of the heat transfer in the different areas of the furnace.

Based on the vaporization heat r, one computes the amount of steam output for every section indicated with m_s.

Similarly to (10.70),

$$m_s = \frac{q}{r}. \tag{10.122}$$

For the i-th generic section the mass flow rate of steam at the inlet M_{s1} is therefore equal to:

$$M_{s1} = \sum_{1}^{i-1} m_s. \tag{10.123}$$

The mass flow rate at the outlet M_{s2} is given by:

$$M_{s2} = \sum_{1}^{i} m_s. \tag{10.124}$$

If the sections are t, the mass flow rate at the outlet of the branch is equal to:

$$M_{se} = \sum_{1}^{t} m_s. \tag{10.125}$$

The values of β_1 and β_2 for the i-th section are given by:

$$\beta_1 = \frac{M_{s1}}{M_{se}}; \tag{10.126}$$

$$\beta_2 = \frac{M_{s2}}{M_{se}}. \tag{10.127}$$

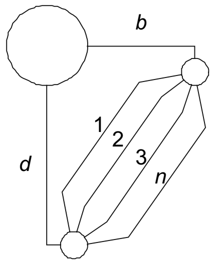
The mean density of the water–steam mix is computed through (10.120) or (10.121). Then it is possible to compute τ through (10.94) where ρ_{mm} obviously corresponds to the value of ρ calculated the way we described.

Subsequently, one computes the mean dynamic viscosity of the mix through (10.95) and the Reynolds number. The value of the friction factor λ is defined based on the value of ε, and the distributed pressure drops in the section are computed by referring, of course, to the value of ρ above.

As far as the concentrated pressure drops at inlet or outlet of the section, one proceeds as usual by considering the value of the density obtained from (10.117), where the value of β is equal to β_1 at the inlet and equal to β_2 at the outlet.

Generally, this is a curve except for the inlet of the first section where $\rho = \rho_w$ and for the outlet of the last section where $\rho = \rho_{me}$. This way one obtains the value of Δp relative to the section in question. So, once the geodetic height difference Δz between inlet and outlet of the section is computed, all quantities included in (10.104) are known. Summing up the factors P relative to the various sections making up the branch in question, one obtains the characteristic factor P of the branch.

Fig. 10.46



Let us consider the generic circuit shown in Fig. 10.46. The downcomers feed the lower collector from which n branches of raisers depart to regroup in one single collector at the outlet. From here the water–steam mix reenters the drum through the return tubes.

Practically, the circuits where the different steam generating branches in parallel regroup directly in the drum are more frequent. The raisers of every branch can also discharge the mix in one specific collector. Then, the mix reenters the drum through return tubes. This situation, though, is different from the one described in Fig. 10.46 where the outlet collector is one for all branches. If every branch has its own collector, from a functional point of view, it is as if the branches discharged directly into the drum. At any rate, when faced with the latter situations, the conclusions we will draw from the study of the schematized circuit in Fig. 10.46 are still completely valid, except for the absence of the return tubes, understood as return branch common to the steam generating branches in parallel.

We indicate d and b as the branches relative to the downcomers and the return tubes and 1, 2, 3,. . .n those relative to the raisers.

Let us consider the circuit consisting of d, 1, and b as the reference circuit.

Similarly to (10.105) we may write that

$$P_d + P_1 + P_b = 0. \tag{10.128}$$

On the other hand, we could write a similar equation for the circuit $d-2-b$, the circuit $d-3-b$, and so on, as well.

Therefore, the implication is that

$$P_1 = P_2 = P_3 ... = P_n. \tag{10.129}$$

Equation (10.129) can also be the direct outcome by observing that $P_1,\ P_2 \ldots P_n$ are nothing but the pressure differences between the inlet and outlet collectors referred to the various branches. They can only be identical. Both (10.128) and (10.129) synthesize the working conditions of the circuit in question.

The calculation process should be as follows. On the premise that the calculation is generally but not necessarily a verification, branch 1 is taken into consideration, a value R_1 of the circulation ratio is set, and P_1 is computed through the process described earlier. Then the calculation is repeated for different values of R_1 to obtain curve 1 shown in Fig. 10.47.

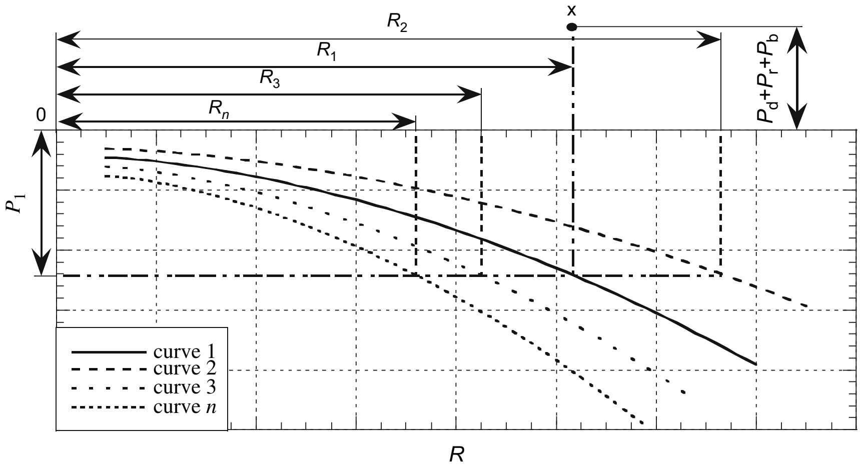


Fig. 10.47 Factor P for various parallel branches as a function of R

The process continues in a similar way for branches $2,3\ldots n$ to obtain the relative curves shown in the figure. At this point, a generic value of R_1 is selected that identifies a value of P_1 through curve 1. According to (10.127), the same value of P must be part of the other steam generating branches. Then, the parallel to the axis of the abscissa passing through P_1 is traced, and the values R_2, $R_3\ldots R_n$ corresponding to the value of R_1 selected in the beginning are identified through the curves $2,3\ldots n$.

If $M_{\mathrm{se1}}, M_{\mathrm{se2}}\ldots M_{\mathrm{se}n}$ stand for the mass flow rates of steam at the outlet of the different branches, and observing that based on (10.71)

$$M_{\mathrm{m}} = RM_{\mathrm{se}}, \tag{10.130}$$

based on the obtained values $R_1, R_2\ldots R_n$, the mass flow rates $M_{\mathrm{m1}}, M_{\mathrm{m2}}\ldots M_{\mathrm{m}n}$ are identified.

The flow rate of water in the downcomers and the flow rate of the water–steam mix in the return tubes is therefore equal to:

$$M_{\mathrm{w}} = M_{\mathrm{mb}} = \sum_1^n M_{\mathrm{m}}. \tag{10.131}$$

The density of the mix ρ_{mb} in the return tubes can be computed as follows.

The mass flow rate of steam M_{sb} is equal to:

$$M_{sb} = \sum_1^n M_{\mathrm{se}} = M_{\mathrm{se}_1} + M_{\mathrm{se}_2}\ldots + M_{\mathrm{se}_n}. \tag{10.132}$$

The water flow rate M_{wb} is equal to:

$$M_{\mathrm{wb}} = M_{\mathrm{mb}} - M_{\mathrm{sb}}. \tag{10.133}$$

The volumetric flow rate of the mix Q_{mb} is therefore equal to:

$$Q_{\mathrm{mb}} = M_{\mathrm{sb}} v_{\mathrm{s}} + (M_{\mathrm{mb}} - M_{\mathrm{sb}})\, v_{\mathrm{w}}. \tag{10.134}$$

The specific volume of the mix is given by:

$$v_{\mathrm{mb}} = \frac{Q_{\mathrm{mb}}}{M_{\mathrm{mb}}} = \frac{M_{\mathrm{sb}} v_{\mathrm{s}} + (M_{\mathrm{mb}} - M_{\mathrm{sb}})\, v_{\mathrm{w}}}{M_{\mathrm{mb}}}, \tag{10.135}$$

or by

$$v_{\mathrm{mb}} = \frac{M_{\mathrm{sb}}}{M_{\mathrm{mb}}} v_{\mathrm{s}} + \left(1 - \frac{M_{\mathrm{sb}}}{M_{\mathrm{mb}}}\right) v_{\mathrm{w}}. \tag{10.136}$$

Assuming that

$$\phi = \frac{M_{sb}}{M_{mb}}, \tag{10.137}$$

then

$$v_{\mathrm{mb}} = \varphi v_{\mathrm{s}} + (1 - \varphi)\, v_{\mathrm{w}}. \tag{10.138}$$

Finally,

$$\rho_{\mathrm{mb}} = \frac{1}{\varphi v_{\mathrm{s}} + (1 - \varphi)\, v_{\mathrm{w}}}. \tag{10.139}$$

Knowing $M_{\mathrm{w}} = M_{\mathrm{mb}}$, ρ_{mb} and the geometric characteristics of the downcomers and the return tubes, it is possible to compute P_{d} as well as P_{b}.

Thus, the value of the term $(P_{\mathrm{d}} + P_{\mathrm{l}} + P_{\mathrm{b}})$ is put on the diagram in Fig. 10.47 in correspondence of R_1. In general, it will not be zero ("x" point of the diagram). The process described for different values of R_1 is repeated. This way, one creates the curve representative of the term on the left hand side of (10.128) (Fig. 10.48). Still based on (10.128), the running condition of the circuit is represented by the point where the curve mentioned above crosses the axis of the abscissa. This point

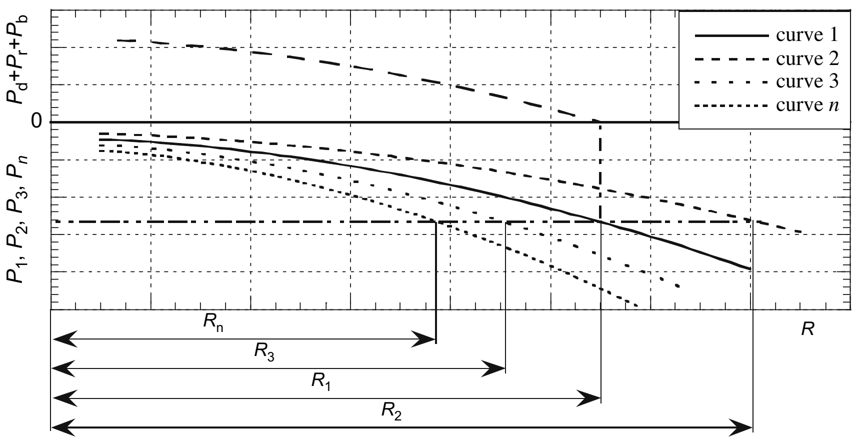


Fig. 10.48 Balance conditions of the circuit

identifies the value of R_1 characteristic of branch 1, and through curve 1 the actual value of P_1 that coincides with the one of the other branches.

The usual process leads to the values of $R_2, R_3 \ldots R_n$ relative to the different branches. At this point one takes the smallest of these values into consideration, and it is possible to judge if the circuit works correctly based on Fig. 10.42. This apparently laborious process is not problematic if the computation is done through a computer. It has repetitive characteristics which are particularly favorable in terms of programming. The calculation of P for the different sections and the subsequent summation to obtain the characteristic factor of every branch can be included in a subroutine called up by the main program as the various branches are examined.

Until now we referred to a verification calculation, given that the dimensional characteristics of the downcomers and of the return tubes are known (the dimensions of the raisers are always set for generators that are already sized from a thermodynamic point of view).

This is the way to proceed during the design phase. The curves relative to the values of $P_1, P_2 \ldots P_n$ of the various branches of the raisers in parallel (Figs. 10.47 and 10.48) must be built even in this case. After determining the minimum value of R through the diagram in Fig. 10.42 based on the running pressure which we indicate with R^*, one identifies the corresponding value of P (it is the greatest in absolute value for the different branches with reference to R^*) with reference to the least favored branch.

Based on this value of P that we indicate with P^* and that must be the same for the various branches in parallel, it is possible to determine the values of $R_1, R_2 \ldots R_n$, as well as the values of M_{w}, M_{mb}, ρ_{mb} through the described process.

In order to ensure the minimum circulation ratio R^* in the least favored branch (in the others the circulation ratio is obviously higher), with reference to (10.128), it must be as follows:

$$P_{\mathrm{d}} + P_{\mathrm{b}} + P^* \geq 0. \tag{10.140}$$

Given that the density of the fluid in the downcomers and in the return tubes is constant, it is possible to examine the corresponding branches in their entirety regardless of the examination of single sections.

Then

$$P_{\mathrm{d}} = \rho_{\mathrm{w}} g \Delta z_{\mathrm{d}} - \Delta p_{\mathrm{d}} \tag{10.141}$$

where Δp_{d} stands for the total pressure drop of the branch. As far as Δz_{d}, it is equal to the height difference between the axis of the drum and the axis of the lower collector at the outlet of the downcomers.

Similarly, for the return tubes

$$P_{\mathrm{b}} = \rho_{\mathrm{mb}} g \Delta z_{\mathrm{b}} - \Delta p_{\mathrm{b}} \tag{10.142}$$

where Δp_{b} stands for the total pressure drop of the branch. Δz_{b} is equal to the geodetic height difference (negative) between the axis of the inlet collector of the return tubes and the axis of the drum.

Based on (10.140) and taking into account (10.141) and (10.142), we have

$$\Delta p_{\mathrm{d}} + \Delta p_{\mathrm{b}} \leq \rho_{\mathrm{w}} g \Delta z_{\mathrm{d}} + \rho_{\mathrm{mb}} g \Delta z_{\mathrm{b}} + P^*. \tag{10.143}$$

The term on the right hand side in (10.143) stands for a known value once the geodetic height of the axis of the drum is defined. At this point the downcomers and the return tubes must be sized by adequately selecting number and diameter in such a way to satisfy (10.143). As the specific volume of the water and steam mix in the return tubes is clearly greater than the one of the water in the downcomers, the reference must be mostly to the return tubes to reduce pressure drops.

The illustrated computational process is the only acceptable one if there are various branches of raisers in parallel. For instance, we consider the circuit in Fig. 10.49 consisting of two branches in parallel of equal length. Furthermore, we assume that they are hit by the same amount of heat and that the only curve has the same dimensional characteristics.

We also assume that the water flowing down the downcomers distributes itself equally in the two branches.

The steam mass flow rate at the outlet of the branches is identical, and based on the assumption that was made this is true for the water–steam mix, as well. The volumetric flow rates are also identical, and so are the two values of ρ_{me}.

Adopting (10.87) the value of ρ_{mm} is also the same in the two branches. On the other hand, the value of Δz_{r} is equally the same, and because of the assumptions so is the value of Δp_{r}.

Therefore, computing P_{r} for the two branches, based on the global values [note the second term on the left hand side of the equal sign in equation (10.103)] and not in reference to the summation of the values of P of the different sections, one obtains two equal values. This agrees with (10.129), and one could draw the conclusion that the circulation ratio in the two branches is actually the same. In reality, nothing could be farther from the truth. The value of R in the left branch is lower than the one in the right branch. The following qualitative points suffice to realize it.

In the rising section of the left branch (first section), the steam starts to form as the water exits from the collector and the density of the water–steam mix decreases

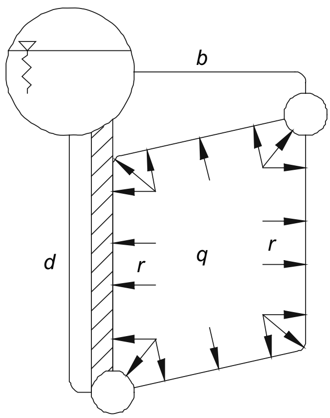


Fig. 10.49

as it moves upward. It is still rich in water, so the mean density in the rising section is relatively high.

In the right branch at the beginning of the rising section (second section), the mix is already rich in steam as a result of the steam generated in the first section. It acquires even more steam while it consequently looses water while it moves upward. The mass flow rate of the water–steam mix being equal, the mean density is therefore lower than the one relative to the rising section on the left (the difference between the two obviously depends on the value of the circulation ratio, which is assumed to be equal in both branches, and is much more sensitive as this ratio is reduced).

The hydrostatic pressure relative to mix column in the left branch is therefore greater than the one in the right branch, while the pressure drops are identical. This implies that the pressure differences between the lower and the higher collector referred to the two branches, that is, the two values of P_r, are not equal. But this is not acceptable, and to reestablish balance a greater amount of water must enter the right branch. This way both the hydrostatic pressure and the pressure drop increase, as well as the absolute value of P_r. Vice versa, the same happens in the left branch in terms of reduced amount of entering water.

The circuit is out of balance with a greater circulation ratio in the right branch, contrary to what the simplistic computation based on global mean values of ρ_{mm} lead to believe. Circulation unbalances in the circuits with parallel branches are quite frequent. Sometimes they are tolerable (if the minimum circulation ratio in the different branches is acceptable), but other times they are not. In that case, it is indispensable to intervene in some way. During the design phase, potential criteria consist of simply improving circulation globally reducing the pressure drops in the downcomers and in the return tubes. In the raisers, interventions of this kind are impossible unless the project is radically modified by adopting larger diameters. This does obviously not eliminate the unbalances, but all the circulation ratios go up which then increases the minimum value in the least fed branch, as well.

Further criteria that involve reducing the flow rate of the water–steam mix in the least fed branches in favor of the one or those poorly fed can be applied either during the design phase or after construction if unbalances are registered. This can be done through throttling plates which increase the pressure drops by reducing circulation in favor of other branches. Of course, global circulation decreases in the sense that the water flow rate in the tubes decreases, but if it is initially abundant, this reduction may not be an obstacle to the insertion of the plates.

Another widespread method consists of planning for balancing valves at the inlet of the different branches (specifically, at the inlet of headers feeding the branches). These valves are tuned as a function of the temperatures registered during runtime on the tubes through thermo-couples.

The same calculation criteria described for natural circulation are true for assisted circulation, too. It suffices to introduce the head of the circulation pump. Equation (10.128) is therefore modified as follows:

$$P_p + P_d + P_l + P_b = 0 \tag{10.144}$$

where P_p stands for head of the pump multiplied by the density of the water and the gravity acceleration.

The value of P_p varies with the flow rate. This will be taken into account by introducing the relative value as a function of M_w computed with (10.131), thus as a function of R_1 based on the described calculation process.

Note that the latter does not include a few aspects of the phenomenon that are worth highlighting. If the steam moves faster than water, the section of the tube ideally occupied by the current of steam is smaller than the assumed one, while it is greater than the one taken up by the water flow. This means that the density of the mix is higher than the one considered up to now.

In the calculations we referred to a constant value of the water-steam density. As far as water, this is substantially correct, but in the case of steam, given the variations in pressure along the raisers, the density actually varies because it is higher than the assumed one from the drum (in fact, the adopted pressure is the drum's) to the lower collector.

The phenomenon is not really important. At the most, the increase in density can amount to 3% from the drum to the lower collector. Thus, on average the increase is 1.5% at the most in the raisers.

Pressure variations in terms of the ensuing spontaneous evaporation are more important. In fact, as the pressure decreases from the collector to the drum, the enthalpy of the water decreases. The heat freed this way generates a certain amount of steam.

The steam evaporated spontaneously may represent up to 7–8% of the steam generated by the heat transfer. Some author states that spontaneous evaporation favors circulation because the density of the water-steam mix decreases. In reality the effect is quite the opposite. Note that water flowing down the downcomers and reaching a pressure higher to that of the drum during the process, stops being saturated water. Once it enters the raisers, a certain amount of transferred heat warms up the water to bring it to evaporation temperature. Then, steam is generated as a result of the heat transfer and spontaneous evaporation. The greater amount of steam generated because of this phenomenon compensates precisely for the steam that was not generated in the initial area of the tubes where the water needed to be heated. It must obviously be this way because at the outlet of the raisers the generated steam cannot be anything but the one corresponding to the transferred heat and the evaporation heat referred to the pressure in the drum, according to the described computation process.

With reference to Fig. 10.50, the density of the water–steam mix behaves like a continuous curve instead of behaving like the dashed curve (theoretic). In the first section (heating of water) the density is practically constant, whereas in the second section (evaporation) the density decreases with a gradient higher than the theoretic one because of spontaneous evaporation.

So, the mean water–steam mix density is higher than the theoretic one, and the impact on circulation is negative. If these phenomena which always impact circulation in greater or smaller negative ways are ignored, there will be insubstantial differences between calculation and the actual phenomenon. Note that the

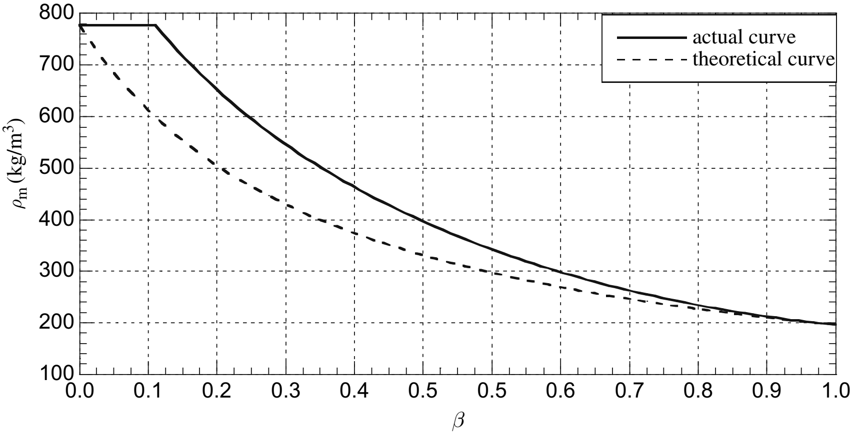


Fig. 10.50 Influence of the hydrostatic pressure and of the spontaneous evaporation

recommended values of R are based on practical experience correlated to theoretical calculation; for this reason, they implicitly factor in these negative phenomena either partially or totally.

In conclusion, we briefly consider a problem of special interest to constructors of generators of small power and pressure. Assuming, for instance, that a generator was designed to work under an absolute pressure of 25 bar and that the circulation ratio is the minimum recommended one, based on Fig. 10.42, the question is whether the same project may be used at lower or higher pressure levels after applying all due modifications to the thickness of the drum, the headers and the tubes.

Figure 10.51 shows the curve of minimum values of R from Fig. 10.42, as well as the values of the circulation ratio for the generator in question, by varying the pressure from 10 to 50 bar. Clearly, by increasing the pressure, the circulation ratio decreases, but it decreases less compared to the minimum allowable values, thus

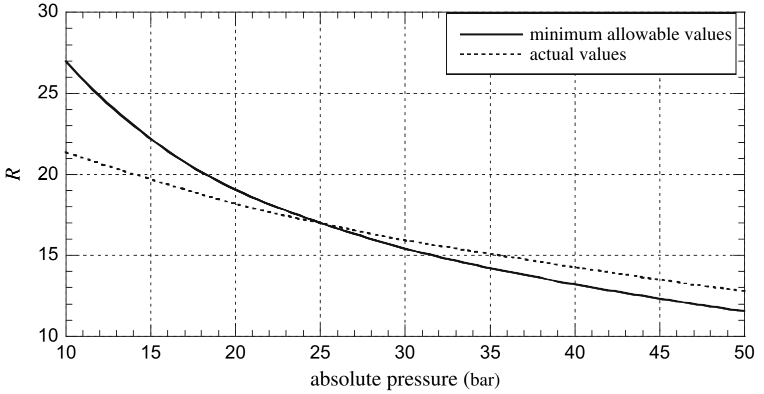


Fig. 10.51

resulting in a higher than minimum ratio at pressures above 25 bar and in a lower than minimum ratio at pressures under 25 bar.

Contrary to common assumption, taking into account the behavioral pattern of the ratio between the density of water and steam at pressure variations, for a given generator low pressure is more dangerous than high pressure.

Therefore, designing a generator that is adaptable to various pressure levels requires planning for runtime under expected minimum pressure and doing computations of circulation based on that.